Vol. XIV. — No. 3. March 1932.

BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

[625. 243 (.92) & 625. 246 (.92)]

All-metal covered wagons for the Dutch East Indies Railway Company (1),

by W. H. van SCHOUWENBURG,
Chief Engineer, Dutch East Indies Railway Company.

General remarks.

The stock of wagons on the « Djocja-Willem I » line required strengthening by 100 covered wagons. This line of 1.067-m. (3 ft.-6 in.) gauge, in general use in Java, branches off our main line, of 1.435-m. (4 ft.-8 1/2 in.) gauge. Under these conditions goods have to be transhipped. From an operating point of view, it is essential the narrow gauge wagons should be of the same capacity as those on the main line on which wagons of 17 tons capacity are in use. During the movement of sugar — the main traffic — the wagons are loaded until nothing more can be got into them.

The narrow-gauge wagons had therefore to be of equal capacity. As the permanent way of the Djocja-Willem I line is lighter than that of the main line it was desirable to have a low tare weight.

Financial considerations.

Whilst taking care that quality should not suffer, an endeavour was made to build light and cheap wagons. To realise these three conditions fully is not easy and it becomes necessary to compromise; to do this the influence of each of these factors on the operating cost of a goods wagon must be ascertained. Only under this condition is it possible to judge with full knowledge of cause if the wagons should be further lightened or not, or if the use of certain improvements is advantageous from a financial point of view.

The operating costs of a goods wagon include :

1. Interest and amortization on the capital invested.
2. Repair costs.
3. Carriage charges.

If we take :

A = Cost of purchase;
a = Rate of amortization;

(1) Translated from the Dutch.

r = Rate of interest;
H = Costs of repairs per kilometre;
k = Mean annual kilometrage;
T = Tare weight in tons;
v = Carriage charges per metric ton-kilometre;

the expenditure on interest and amortization $= A \times \frac{a+r}{100}$;

the annual cost of repairs $= H \times k$, and the carriage charges $= T \times v \times k$, so that the total operating cost

$$E = A \times \frac{a+r}{100} + H \times k + T \times v \times k.$$

In this formula, a, r, k, and v, may be considered as constants seeing they do not affect the type of wagon. Consequently the values A, H, and T, directly depend upon the materials used and upon the type of wagon.

In a carefully enquired into case, we found the following :

$\frac{a+r}{100}$. — The annual rate $\frac{a+r}{100}$, corresponding to rates of interest of 4 % or of 5 %, calculated for a theoretical period of repayment of 15, 20, 25, and 30 years, has the value of :

RATE OF INTEREST.	Period of repayment :			
	15 years.	20 years.	25 years.	30 years.
4 %	8.99	7.36	6.40	5.78
5 %	9 63	8.02	7 09	6.50

A repayment period of 25 years with a 5 % rate of interest was selected.

H. — The cost of repairs of the existing goods wagons amount to 0.75 cent per wagon-kilometre (1.2 cents per wagon-mile) for four-wheeled wagons.

k. — The average annual kilometrage is 12 000 (7 456 miles) for the 4-wheeled wagons.

T. — The tare of the new 17-ton wagons was 7 tons in round figures.

v. — The carriage charges only include the direct charges relating to the gross ton-kilometre transported. These costs, therefore, are not the cost prices. The latter include a part proportional to the whole of the charges for management, administration, maintenance of the permanent way, repairs of rolling stock (already mentioned under heading H), etc., etc.

The carriage charges we are considering here include exclusively the cost of haulage, that is to say the fuel, the water and oil used. These amount in a case investigated to 0.1 cent per gross ton-kilometre (0.16 cent per gross ton-mile).

Under such conditions the purchase price of a wagon being taken as 3 500 florins, the annual operating cost of a wagon becomes :

Interest and sinking fund	$= 3\,500 \times 7.09\ \% =$ Fl. 248;
Repairs	$= 12\,000 \times 0.0075 =$ Fl. 90;
Carriage charges	$= 7 \times 0.001 \times 12\,000 =$ Fl. 84.

We can therefore say that in the case under consideration the financial charges relating to the capital cost represent about 60 % of the operating costs, whereas the costs of repairs and carriage only represent about 20 %. Moreover these considerations must not be taken as applying to new stock alone.

For this reason most railway companies divide the maintenance costs into the costs of repairs to the wheels and axles, axleboxes, springs, draw and buffer gear, and to the frame and the body. As a result of subdividing the charges in

this way, it is possible to judge if any given alteration made to a type of wagon is beneficial or otherwise from an economic aspect. If, for example, an alteration in design costing 60 florins involves an increase in weight of 40 kgr., in this case the first item will increase by 4.2 florins annually and the third by $0.04 \times 0.001 \times 12\,000 = 0.48$ florin, which in all represents an increase of 4.68 florins, so that the change will be justified if as a result the repair costs are reduced by at least 5 %.

In addition the age of the wagons must be taken into account. If they are already getting old, the expenditure involved by the alterations — when the life of the wagon remains unaltered — must be amortized quickly, so that the amortization rate will be higher. Consequently if, by careful design, it were possible to save 200 kgr. on the weight of a wagon, so as to make it cost less say by 50 florins, in this case the first item would be reduced by 3.50 florins and the third by $0.2 \times 0.001 \times 12\,000 = 2.4$ florins, that is a total of about 6 florins.

These considerations were taken into account when building the new wagons. None the less it is customary to increase the stock by buying more wagons of an existing type modified if necessary. In the present case, however, the Dutch Indies Railway Company introduced an entirely new type with the result that a check had to be made at different times to make sure that all constructional details were justified.

Construction.

The forces stressing a structure affect the dimensions selected. These forces are known in the case of a stationary wagon, and we also know approximately the stresses acting on a wagon in motion; but we are not too well fixed as to the importance of the shocks a wagon undergoes during shunting. The regulations generally require shunting to be done at walking pace, but in practice this is far from the case. It is in fact not unusual for the wagons to run into one another at about double this speed and, however contrary to regulations this may appear, it is not exaggerated to require the wagon to be capable of standing such shocks without being damaged.

The momentum of a fully-loaded 17-ton wagon of 7 1/2 tons tare at the moment of a shock of this kind is equal to

$$\tfrac{1}{2}mv^2 = \tfrac{1}{2}\,\frac{7\,500 + 17\,500}{10} \times 3^2 = \sim 11\,000$$

Kg.-M. (79 563 foot-pounds).

A buffing spring compressed 7 cm. (2 3/4 inches) offering a uniformly increasing resistance up to 15 tons can absorb a maximum amount of energy of $\frac{15\,000}{2}$ kgr. $\times$ 0.07 m. = 525 Kg.-M. (3 797 foot-pounds). The wagons have centre drawgear without side buffers. When a moving wagon runs into a standing one, a maximum of $2 \times 525 = 1\,050$ kgr. (7 594 foot-pounds) is absorbed in this way. After the buffers are compressed solid there still remains about 10 000 Kg.-M. (72 330 foot-pounds to be destroyed. This destruction of kinetic energy takes place by change in speed of the wagons coming into contact, by displacement of the load, and by deformation of the wagon.

Up to the present it has been found impossible to calculate each of these parts. If we remember however that there are wagons on the 4 ft.-8 1/2 in. gauge which, when fully loaded, weigh almost double those in question and are fitted with two buffers each able to absorb 30 tons, that in addition, at equal

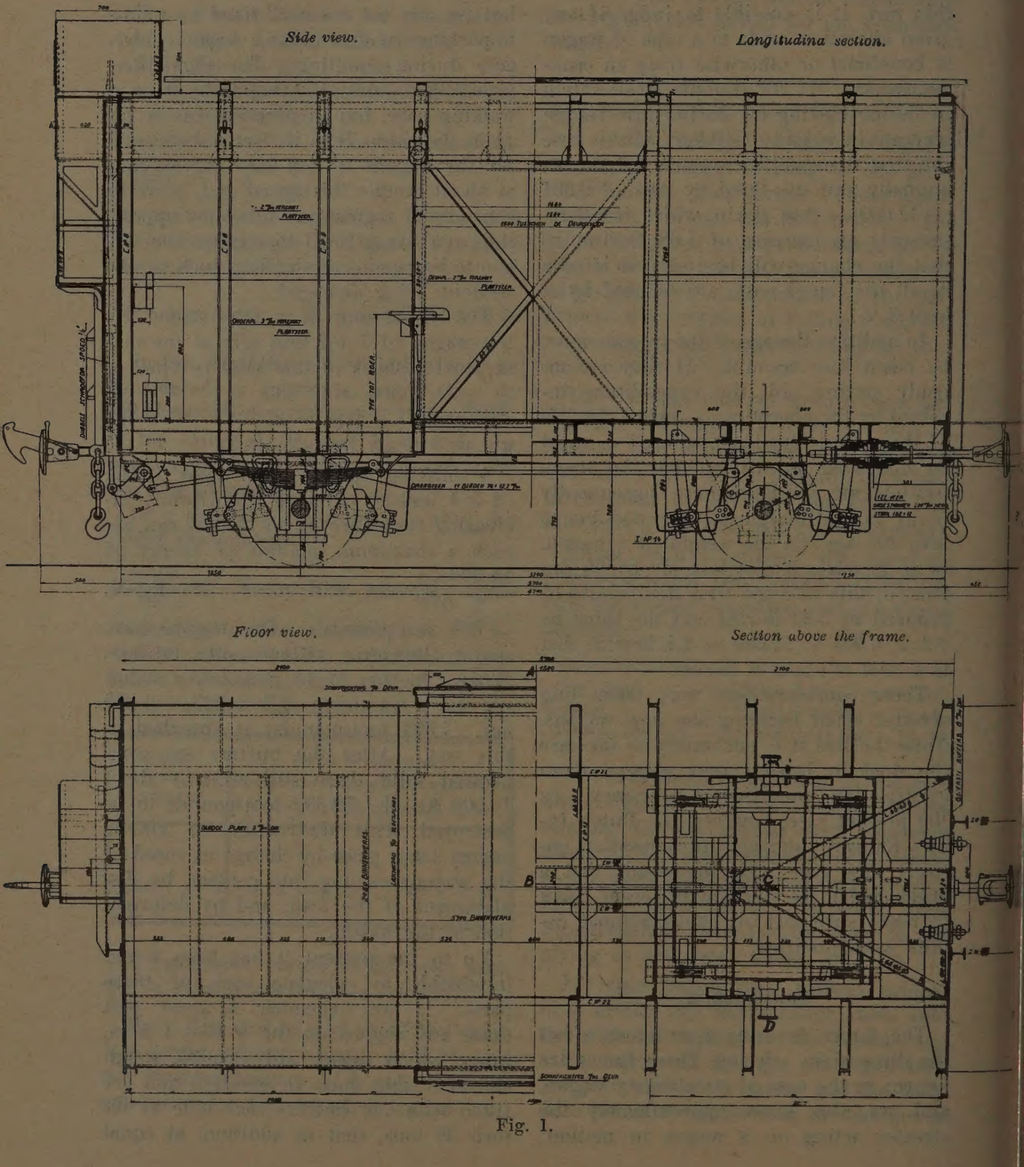

Side view.
Longitudina section.
Floor view.
Section above the frame.

Fig. 1.

speed, the kinetic energy is proportional to the weight, it appears reasonable to admit that the force acting between the wagons may fall to a half of 2 × 30 tons = 30 tons.

On the basis of this hypothesis the wagon design of the Department of the Indies was checked and then built by the « Waggonfabrik Uerdingen » who took every care in its construction. This firm put forward certain alterations to the frame with a view to improving the way the forces acted without moreover increasing the weight thereby. It also took the greatest interest in the tests to which the wagon was subjected subsequently and thereby largely contributed towards the good results obtained.

Figure 1 gives the general arrangement of the wagon of which the following points deserve especial attention.

Frame.

The frame consists of a double central longitude of U section iron of standard profile 14.5 (5 45/64 inches), which has to absorb the draw and buffing stresses. At right angles, at distances varying from 480 to 540 mm. (18 7/8 to 21 5/16 inches) there are bearers of U section iron, standard profile 12 (4 3/4 inches), which carry the floor of 5-mm. (3/16 inch) plate, the outer ends of these bearers being connected together and to the buffer beams which consist of U section of No. 22 (8 11/16 inches) standard profile. There was some possibility that, as a result of a violent shock, the bearers might be deformed as shown in figure 2. To prevent this, the centre longitudes were connected to the headstock by two diagonals (see fig. 3). The longitudinal movement of the centre longitude was thereby prevented.

Body.

The body is composed of U-shaped iron pillars of standard profile No. 8 (3 1/8 inches) connected together longitudinally 1.75 m. (5 ft. 9 in) above floor level by a Z bar. In previous types the body was lined out with wood, which absorbed the whole of the stresses due to the load. The wagon was covered outside with galvanised iron sheets which protected the body against the weather.

The outer sheets were held by wood screws. With this method of fastening ultimately the screws slackened and then water got in. For these reasons the wagon described below was built all metal, no wood at all being used. The galvanised-sheet sides are rivetted to the uprights. The roof rests on the sides and is held by tightening straps (see fig. 4).

Special care was taken to get a water-tight body even in the torrential rains of the tropics. The roof is built of sheets put in place cold, free one against the other, over the No. 8 standard U profile sections. They are held tightly down on these U sections by clamps tightened by bolts. The roof sheets therefore are not drilled; any possible drops of water which may get in between the clamps and the roof are collected and taken away in the trough of the U section. This form of construction is a good one and has given complete satisfaction. It cannot however be used over the door opening because the U section bars stop in line with the Z bars at the cornice. Water might get in at this point through the door. For this reason over the full door opening a single roof sheet 1 584 mm. (5 ft. 2 3/8 in.) wide has been used and has been rivetted to the vertical flange of the Z bar. The door is carried on rollers which run on the flat part of the Z bar (see fig. 5).

The side walls top half is made of 2-mm. (5/64 inch) sheet and the lower half of 3-mm. (1/8 inch); they overlap one another and are rivetted at the joint (see fig. 4). The bottom plates are carried 70 mm. (2 3/4 inches) below the flooring so as to prevent any water getting in.

To meet any possible movement of the load through shock, the ends have been reinforced by two I bars with wide flanges, of 100 × 85 mm. (3 15/16 × 3 3/8 inches) sections. The ends are rivetted to the flanges of these I bars (see fig. 6).

The spacing of the rivets securing the sides to the ends was 8 cm.(3 1/8 inches), but to make the joint tighter against rain it is desirable to reduce it to 6 cm. (2 3/8

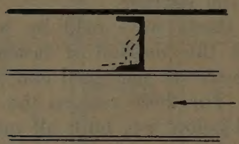


Fig. 2.

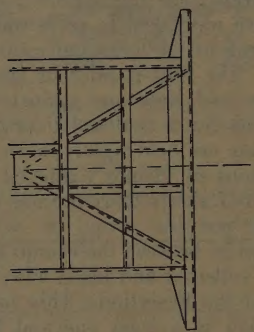


Fig. 3.

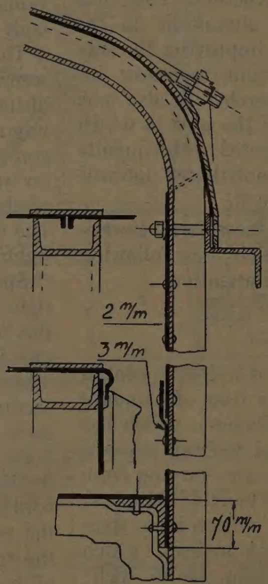


Fig. 4.

Fig. 5.

Fig. 6.

inches) in the case of sides of the thinness used.

Figure 7 gives a perspective view of a corner of the wagon body.

Special attachments fitted for test to some of the wagons.

« Armco iron » sheets for roofs and sides on 10 wagons.

Springs to brake blocks on all the 60 braked wagons.

« Uerdingen » ring springs to 5 wagons.

« van Leer » brake gear to 2 wagons.

« *Armco iron* » *sheets*. — On 10 wagons the roofs and sides were built of galvanised « Armco iron » sheets which involved an additional expenditure of

33 florins per wagons. This expenditure will be paid off provided the cost of maintenance of these sheets is reduced by 33 × 7 % = 2.30 florins per year. Experience alone will show if this will be the case.

Brake block springs. — In order to prevent the brake blocks dragging on the wheels, tempered springs of the design shown in figure 8 were used. The spring is passed through an extension of the hanger and sets up sufficient friction to keep the brake block in position. The method is simple and, from reports received from the Indies, appears to give good results.

« *Uerdingen* » *ring springs.* — As will be seen from the general arrangement drawing, the buffer and draw gear were of a primitive design. The principle of construction in accordance with which a spring is fully unloaded both in traction as under shock can give but little satisfaction. The great advantage of the design lies in its simplicity and cheapness.

The « Uerdingen Waggonfabrik » put forward a proposal to fit several wagons with the « Uerdingen » ring springs. Finally five wagons were fitted (see fig. 9).

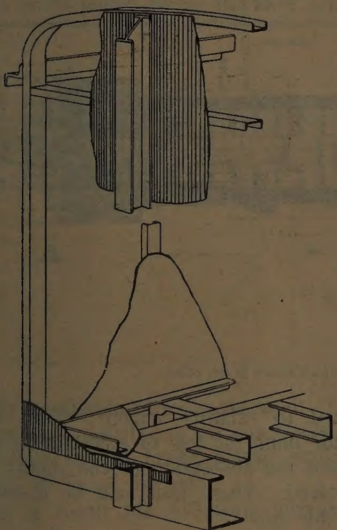


Fig. 7.

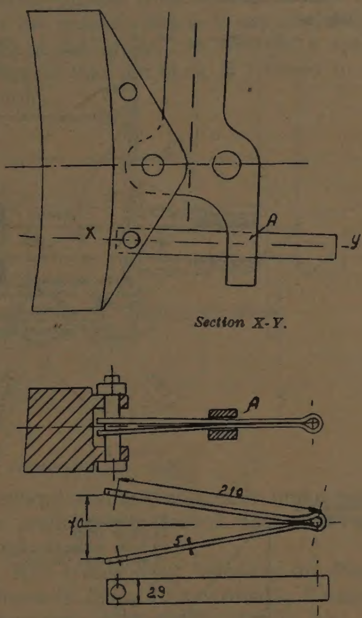


Fig. 8.

The « Uerdingen » ring spring, as is known, is a friction spring which absorbs a considerable amount of work and largely annihilates it so that only a small proportion is given out on the rebound. The work diagram of the spring made is given in figure 10.

Amongst the advantages of the system

we should mention that in comparison with a volute spring and with the same kinetic energy, a greater part of this energy is absorbed by the buffing gear, so that the wagon frame is less liable to be subjected to direct shocks and so may be made lighter.

The extra cost of purchasing this type of buffing and draw gear can therefore be balanced to some extent by using a lighter construction, and one therefore less costly.

The use of the « Uerdingen » ring springs would appear to make it possible to get a different and better distribution of the materials in the wagon.

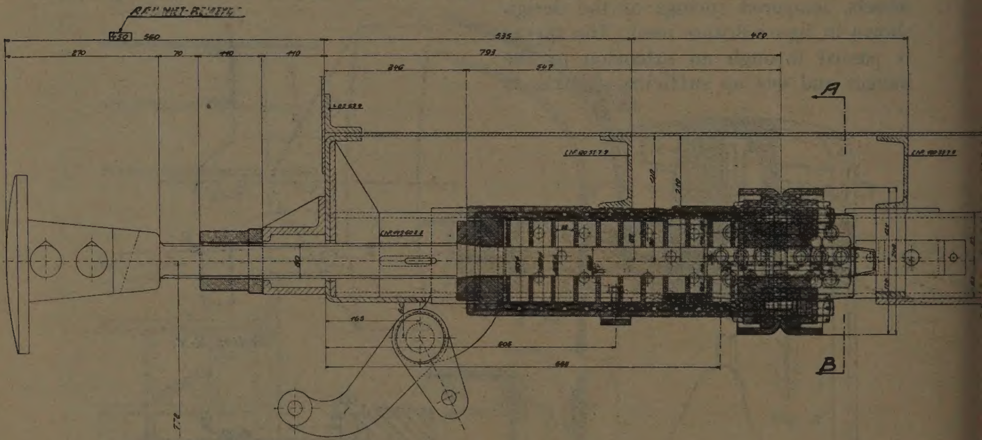


Fig. 9.

Note : Aan niet-remzijde = End at which brake is not fitted.

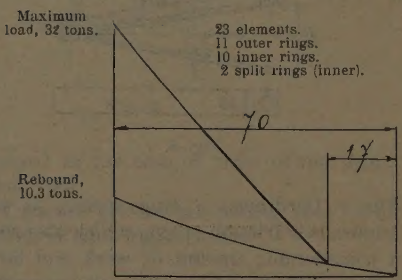


Fig. 10.

These supplementary advantages evidently cannot be obtained in the present wagon, seeing that the ring springs were fitted to existing wagons. Tests were made none the less to ascertain to what extent the absorption of the shocks was bettered. These tests will be described further on.

van Leer patented brake gear. — The brakes in general use suffer from the drawback that the brake power is independent of the degree of loading of the wagon. If the brake power be 2/3rd of the weight of the loaded wagon, in the event of the wagon being partially loaded or empty, the wheels may be picked up. If on the other hand the brake power is so reduced that there is no danger of the

wheels being skidded, the total brake power for the same number of brakes in use is so low that it is not possible to count on getting a good enough braking action.

Many efforts have been made to remedy this defect by adding automatic devices.

The patent which has been applied in the present case (see fig. 11) makes use of the flexion of springs subjected to the action of the load. This flexion is transmitted, amplified, to a block free to move in a guide which forms the lever arm of the brake gear. For each position of the spring there is consequently a corresponding position of the block and a well-defined lever arm ratio.

The operation of the brake has the effect of wedging the block in the guide, which results in a braking force proportional to the ratio at that moment of the arms of the brake lever of the guide, a ratio resulting from the degree of flexion of the springs, that is to say of the load. When the brakes are released a spring ensures that the block is released in the guide.

Empty : $R = K \times \frac{A\,B}{B\,C}$.

Loaded : $R' = K \times \frac{A'\,B'}{B'\,C'}$.

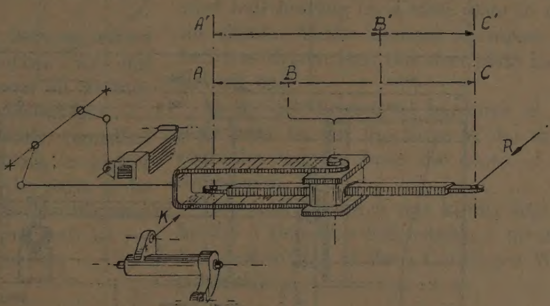


Fig. 11.

By a suitable ratio of the arms of the lever, it is possible to get constantly and automatically a brake power equal to about 90 % of the load.

As a result the following advantages are obtained :

1. The skidding of the wheels of the wagon when empty is avoided;

2. At all times the maximum brake power allowed by the conditions can be utilised;

3. The stopping length is proportionally reduced;

4. If all wagons are equipped with this brake, the number of brakesmen can be reduced, seeing that greater brake power is available per loaded wagon and per brakesman.

This arrangement following the design shown in figure 12 was fitted to 2 wagons for trial. The results of this trial are given below.

Tests carried out.

Amongst the 100 wagons ordered, two were built first, fitted with the brake, and subjected to the following tests :

1. Test as to tightness against rain;

2. Test under shock to measure the value thereof and to check the strength

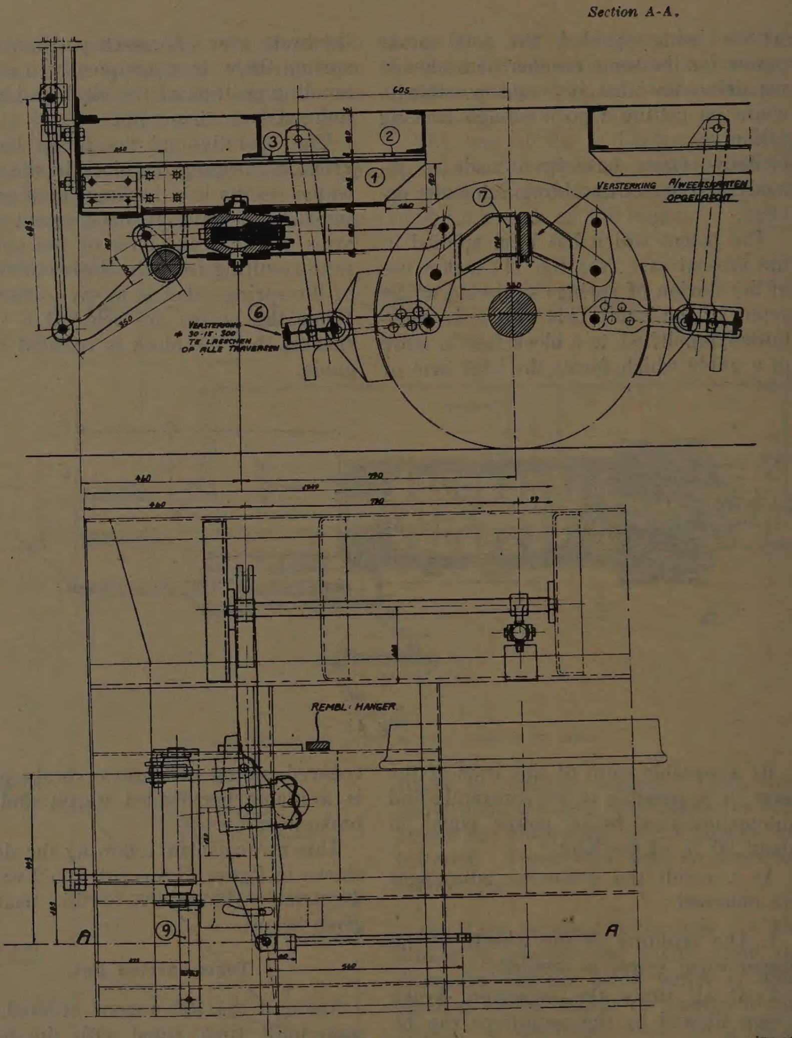


Fig. 12.

of the frame and of the body, chiefly as regards the headstocks and the diagonals;

3. Test under shock of the « Uerdingen » ring springs in comparison with ordinary volute springs;

4. Brake tests with the « van Leer »

patented fitting in comparison with the ordinary hand brake.

1. — **Test for tightness against rain.**

An empty wagon had a jet of water thrown onto it from a fire nozzle. The jet was directed both vertically and horizontally on to the wagon so as to imitate heavy tropical rain. The first tests showed some infiltration of water along the vertical joints and all along the upper joint of the doorway. It was found possible to cure the former, but in the case of the latter it was necessary to build the roofing above the doorway in one piece as mentioned earlier on.

In service the distance between rivets fixed at 80 mm. (3 1/8 inches) was not close enough to exclude under all circumstances all risk of water getting in.

This defect was completely remedied by inserting, before rivetting, a plastic jointing material between the parts to be rivetted. For future wagons the distance between centres of the rivets will be reduced to 60 mm. (2 3/8 inches).

2. — **Shock test.**

A line with at one end a section running up a sharp gradient was laid down in the Work's yard. By pushing one of the two wagons more or less far up the slope, the wagon could be given a more or less high speed. Experience showed the position the wagon had to occupy on the slope for it to attain a speed of 6, 8, or 10 km. (3.7, 5 or 6.2 miles) an hour on the level after being released. These points were marked in white on the slope. The other wagon was placed on the level stretch so that the moving wagon, when coming on to the level, would run into it at its maximum speed. The exact speed was recorded by a tachymeter watch. The tests were carried out using loaded wagons, the load consisting of sacks of sugar. It was found, on this occasion, that the proper number of sacks could be loaded. The stationary wagon subjected to the shock tests had its brakes on in some cases and off in others.

The object of the tests were :

a) to calculate the force of the shock;

b) to observe the deformation of the headstocks subjected to these shocks.

a) *Determination of the force of the shock.* — The wagons were fitted with centre buffers. On the wagon which had to run down the slope, the buffers were replaced by a box containing a hardened-steel ball bearing on a steel plate to take the shock. At the moment of impact the ball was driven into the steel plate like a Brinell ball.

As the coefficients of hardness of the plate were known, the force of the shock could be deduced from the depth of the impression.

This testing device was kindly lent us by the « Deutsche Reichsbahn » through the good offices of the « Uerdingen Waggonfabrik ».

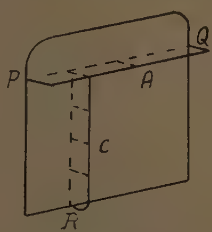


Fig. 13.

b) *Observations taken of the deformation of the headstocks.* — The forces acting on the wagon ends through movement of the contents, are taken by two I section end studs (see fig. 13) connected together by a cross-piece.

The sheets of the sides, which are rivetted to the pillars, form a longitudinal

assembly of such rigidity that the corner pillars may be considered as fixed points of support. The same thing occurs with the headstocks. For these reasons a member is secured at P, Q, and R as shown in figure 13, whilst friction blocks take a bearing at A and C, these blocks moving through friction and remaining in the position to which they have been moved.

Results of tests with volute springs (see table I).

These results tend to indicate the effect of the method of securing the load in position. The better the load is secured the less it can move, and the less momentum it can absorb. The result is the frame has to absorb more.

TABLE I.

Speed in kilometres per hour.	Wagon standing brakes *on* or brakes *off*.	Impression of the ball in millimetres.	Maximum value of the shock in metric tons.	Deflection in millimetres. Momentary. A*	C*	Permanent. A	C*	*Remarks.* (*) See fig. 13.
11.5	O	15.5	57.7	2.5	2	...	...	Under the action of these first shocks, the load was pushed to one side.
4 8	Off	8	15.1	0.5	0.5	...	...	
5.7	On	10.5	26.1	1	1	...	...	
6.4	Off	10.6	26 5	1.5	1.5	...	..	
9.6	On	15.4	56.8	5	4	...	...	The load is now secured against the leading end and consequently only takes a little of the work due to the shock.
7.2	On	15	53.9	4.5	4	...	...	
9.6	Off	17.1	71.2	8	6	...	...	
9.6	On	17.3	72.4	9	7	1	1.5	
(1)7.2	Off	14.35	49.3	6	4.5	0.5	0.5	(1) The tachymeter watch was not correctly used so that the speeds are inaccurate.
7.2	On	11.95	33.7	6	5	0.5	0.5	
5.76	Off	11.8	32 8	2	1.5	...	. .	
5.76	On	11.3	30.3	2.5	2	...	...	
11.5	Off	17.65	74.8	10.5	8	1.5	0.5	
11.5	On	17 9	78 3	11.5	8	1.5	1	After this shock, the buffer spindles were bent.

During the last two blows which occurred at the same speed as the first, the shock was appreciably higher (about 75 tons) than at the first (about 58 tons). During the last blows, in addition to the bending of the buffer spindles, there was a permanent set of the end pillars [1.5 mm. (0.06 inch)] from which we must conclude that the stiffness of these pillars is just ample, although not excessively so.

Furthermore these tests show that the stress of 30 tons, originally considered the maximum wagons had to bear, is very appreciably exceeded in the shocks which occur during shunting and which are not infrequently met with in practice. It therefore became necessary to examine

Graphical representation of the shock caused by a moving loaded wagon running into a stationary loaded wagon.

Stationary wagon, *brake off.*

The two wagons fitted with a *volute spring.*

Tare weight	7.3 tons.
Load	17.85 »
Total weight . . .	25.15 »

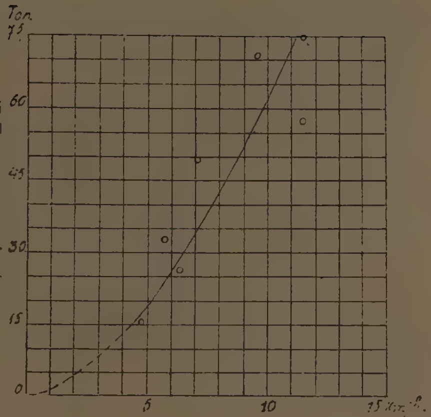


Fig. 14.

Graphical representation of the shock caused by moving loaded wagon running into a stationary loaded wagon.

Stationary wagon, *brake on.*

The two wagons fitted with a *volute spring.*

Tare weight	7.3 tons.
Load	17.85 »
Total weight . . .	25.15 »

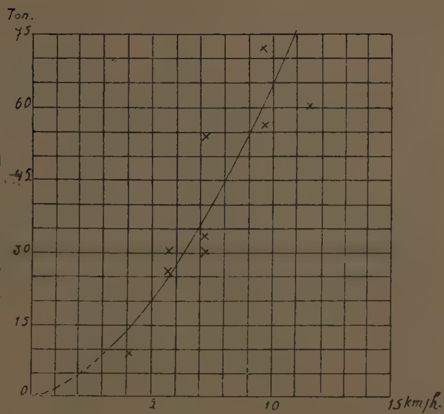


Fig. 15.

the wagons to make sure that as a result of these severe shocks they had not been damaged.

To check this the rivets of the frame were hammer-tested after each severe blow, but no loose rivets were found.

This favourable, though negative, result still did not solve the question as to whether the diagonal braces, provided to prevent any movement of the centre longitudes under severe shocks, were indeed necessary. In order to clear up this point, the rivets holding these braces were cut out on one side of the wagon and re-

placed by bolts with a little play in the holes (3 holes of 17 mm. (11/16 inch) in which 5/8-inch bolts were used).

After three blows of 50, 60, and 70 tons respectively the bolts bent. The rivetted braces therefore fulfilled the duty expected of them. They resisted any longitudinal displacement of the centre longitude and thereby prevented the cross bearers from getting out of shape which ultimately would mean costly repairs.

Figure 14 shows graphically the results obtained with a standing wagon with the brakes off and figure 15 the results obtained with the same wagon with the brakes on.

These diagrams show that there is very little difference between the effects of a blow on a wagon with the brakes on and on a wagon with them off. The blow occurs at such a high rate that the fact of the brakes being on has no effect on the wagon. Naturally the wagon runs much further when the brakes are off.

3. — Shock tests repeated this time with the « Uerdingen » ring spring.

The violent shocks to which the frame was subjected led the « Uerdingen Waggonfabrik » to consider the question as to whether by replacing the volute spring by a ring spring it would be possible to absorb a greater proportion of the momentum and thereby reduce the part of the momentum not absorbed acting on the frame. Both of the wagons subjected to the test were fitted on one end only with ring spring gear and were run into one another so that the ends so fitted came into contact.

The table below gives the results obtained with these ring springs.

TABLE II.

Speed in kilometres.	Wagon at rest with brakes on or off.	Impression of ball, in millimetres.	Maximum value of shock in metric tons.	*Remarks.*
5.5	Off.	9.66	21.9	Wagons loaded with sacks of sugar.
7.2	Off.	9.94	23.6	
11.5	On.	12.72	38.8	
13	On.	13.76	45.6	
6.37	Off.	7.5	13.4	
5.47	Off.	8.65	17.6	
5.47	On.	9 10	19 7	Body removed. Frame loaded with billets.
6 01	On.	9.35	20.5	
10 94	Off.	14.6	50.6	
10.94	Off.	16.6	65.8	
6.84	Off.	10.15	24.3	
6.37	On.	10.85	28.1	

Comparison between the « Uerdingen » and the volute springs.

The wagons fitted at one end with the « Uerdingen » springs were turned end for end after having received three blows; the volute springs were then subjected to as near as possible the same shocks.

TABLE III. — **Comparative results obtained with the « Uerdingen » ring springs and the volute springs.**

Speed in kilometres.	Wagon at rest with brakes on or off.		Impression of ball in millimetres.	Maximum value of shock in metric tons.*	*Remarks.*
6	On	ring spring.	7.9	11.3	The load of the wagon consisted of billets loaded on the frame.
7.2	On		10	23.4	
11.52	On		15.2	55 3	
4.1	On	volute spring.	6.8	8.8	
7.2	On		11 4	30.5	
11.52	On		15.8	60.3	

Figure 16 shows graphically the results obtained with the « Uerdingen » ring spring, the wagon having the brakes off, and figure 17 those obtained when the brakes were on.

Here again the tests show that the value

Graphical representation of the shock caused by a moving loaded wagon running into a stationary loaded wagon.

Stationary wagon, *brake off.*
The two wagons fitted with a « *Uerdingen* » *ring spring.*

Tare weight	7.3 tons.
Load	17.85 »
Total weight . . .	25.15 »

Note. — In the case of the tests marked with a cross, the speed was not taken with a tachymeter watch.

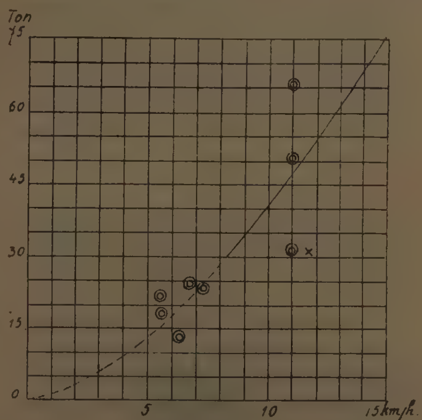


Fig. 16.

of the shock on the wagon at rest is only slightly affected by the brakes being on or off.

If, when examining the graph, figure 18, we compare the stresses due to the blow absorbed by the « Uerdingen » spring with those absorbed by the volute spring we see that the « Uerdingen » springs absorb a greater part of the work and that as a result the frame is less

Graphical representation of the shock caused by a moving loaded wagon running into a stationary loaded wagon.

Stationary wagon, *brake on.*
The two wagons fitted with a « *Uerdingen* » *ring spring.*

Tare weight	7.3	tons.
Load	17.85	»
Total weight . . .	25.15	»

Note. — In the case of the tests marked with a cross, the speed was not taken with a tachymeter watch.

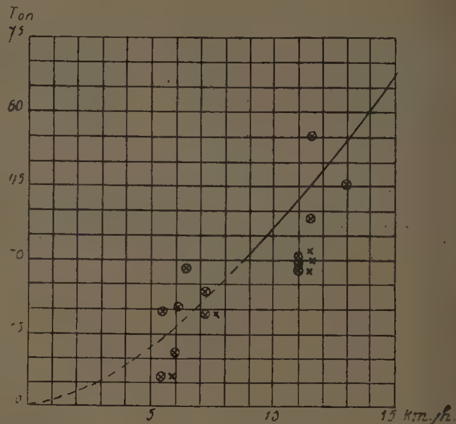


Fig. 17.

subject to the action of the stresses not absorbed. In the case of a blow at 8 km. (5 miles) an hour, the frame with ring springs is subjected to a shock of 30 tons whereas, when fitted with volute springs, the shock reaches 45 tons.

At the speed of 10 km. (6.2 miles) an hour, these values become 40 and 60 tons respectively.

4. — Brake test.

Braking effort. — With the object of obtaining always comparable braking power, the brakes were applied by means of weights.

Stopping length. — The brake was applied by releasing a tackle operated by a trip in the four-foot. The stopping length was measured from this trip to the point at which the wagon stopped.

As for the same brake block movement

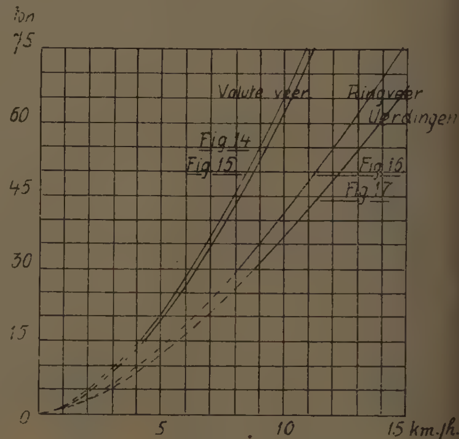


Fig. 18.

Explanation of Dutch terms:
Volute veer = Volute spring. — Ringveer Uerdingen = Uerdingen ring spring.

the number of turns of the brake screw is proportional to the brake gear ratio, the time taken for the blocks to be applied to the wheels is longer when the leverage is great than when it is low. From this fact the length of the stop will increase somewhat with increasing loads, although the force of application of the brake blocks increases in proportion to the load.

Speed. — The wagon to be braked was pushed each time to the same position on the hump from which it ran freely so that — in spite of the influence of the load — the speed, measured at the trip, was the same in each case.

This speed was as high as 15 km. (9.3 miles) an hour.

Load. — The test was made with the wagon empty and loaded with 6, with 12, and with 18 tons.

TABLE IV. — **Stopping distances of a wagon fitted with the "van Leer" patented brake and carrying different loads.**

Load of wagon.	Stopping length in metres (*in feet*).	Average distance in metres (*in feet*).	*Remarks.*
Empty	10 (32.8)	9.83 (32.24)	...
do.	10 (32.8)		
do.	9.5 (31.2)		
6 tons.	11 (36.1)	10.6 (34.77)	...
do.	10.1 (33.1)		
do.	10.7 (35.1)		
12 tons.	10.7 (35.1)	10.3 (33.8)	...
do.	10.2 (33.5)		
do.	10.1 (33.1)		
18 tons.	11 (36.1)	11.4 (37.4)	...
do.	12.1 (39.7)		
do.	11.1 (36.4)		

The practically equal values of the stopping distances, although with very different loads show that the retarding force and consequently the ratio of the brake power to the weight are sensibly the same; in other words, that the brake power increases with the load.

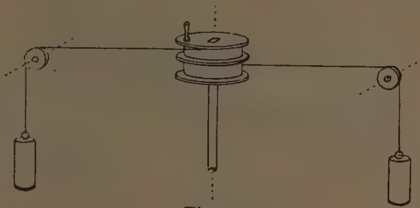


Fig. 19.

For comparison, the van Leer fittings were put out of action when the wagon was most heavily loaded, and the wagon, now fitted with an ordinary brake, was braked with the same counterweights. The stopping distance increased to 25 m. (82 1/4 feet) as a result.

* * *

The above tests were repeated in Java and gave the same results. The wagons were then put into service without anything more being done to them, and have given rise to no difficulties after several months use.

To sum up, we can say that we have succeeded in making a wagon of sufficient strength in view of the results of the tests to which it was subjected. Naturally, we have no sufficient data as regards maintenance costs.

However, the wagon is a light one. It has a capacity of 17 tons for a tare of 6.6 tons for the unbraked wagon and 7.3 for those brake-fitted, which equals 39 % and 43 % respectively of the loading capacity.

The costs of these wagons are not high when compared with those of other wagons. The costs in the case of 15-ton wagons which were already in service amount to 3 500 florins for the unfitted wagons, and 4 100 florins for the brake-fitted wagons, whereas the new unfitted wagons cost 3 250 florins and the fitted 3 600 florins, taking into account all expenses on the Java lines.

Seeing that — other things being equal — a new type is always more expensive than an existing one, we may expect lower prices when placing further orders.

[385 .587 (.493) & 656 .212.5 (.493)]

Application of the "belt system" to the Belgian National Railway Company's Antwerp-North marshalling yard,

By F. DESSENT,
Chief Engineer, Operating Department, Belgian National Railway Company,
and J. COLLE,
Engineer, Operating Department, Belgian National Railway Company.

In a marshalling yard work is done to which the « belt system » can be applied.

All trains entering the yard have to be shunted so as to sort out the wagons, first of all by direction and then to form them in the geographical order of their destinations (see appendices 1 and 2).

The result is that the deciding element in the output of a marshalling yard is the cadence of the successive shunts, that is to say the speed the rakes are pushed up the humps, the interval between two pushes moreover being reduced to the minimum.

All the other operations both those preceding and those following the shunting should be carried out in a period of time determined by the cadence of the shunting itself.

At Antwerp North the belt system has been made to cover all work done to a train from the moment it is received in the yard until the last vehicle has been shunted.

When considering the belt system the investigation should therefore bring every operation (or group of operations carried out at the same time) into a period of time such that the cadence of shunting should not be slowed down. In other words, care must be taken to prevent some gangs having to wait because preceding gangs have not finished.

To ensure this either certain gangs must be strengthened or others reduced in numbers so as to get the desired rhythm of working over the entire system.

Operations in the C_1 group of reception sidings.

The trains are received in the C_1 group of sidings at Antwerp-North, and stand there waiting to be shunted (appendix 1).

The time the wagons are in the reception sidings includes :

1. Air brake examination, inspection of stock, clerical work in connection with the train, preparatory work prior to

shunting including drawing up the shunting list, deciding the cuts to be made, releasing the air brake, and lastly the completion and distribution of the shunting and braking lists.

2. Pushing up over the hump and the actual shunting.

The detailed analysis of the various operations carried out with the train in the reception sidings was undertaken in order to get proper coordination between the men of the different departments who take part in the shunting work.

The analysis began by taking a large number of stop watch timings of the length of time of the work done by all the men and also of the movements of the shunting engines.

The result was that the average times a man required for dealing with a 60-wagon train involving on the average thirty cuts are as follows :

A. — Preparation of the train for shunting.

1. *Clerical work.*

	Minutes.
a) Work at the train, a clerk :	
Walking down the train, collecting the papers, marking the wagons, checking the seals and loads. .	40
b) Office work, a clerk :	
Stamping and sorting the bills by destinations (3 min.) ; preparing the advice notes of the dispatch of the bills (6 min.) ; abstracting the bills according to the different destinations (8 min.) ; completing the train sheet (7 min.) ; correcting any errors found (6 min.)	30
That is, for dealing with the train from a clerical point of view. .	70

2. *Examination of the train.*

	Minutes
a) Air brake examination (one examiner) :	
Testing for leakage and correct working	18
b) Examination of stock :	
Technical examination of the wagons as regards damage : two examiners (with the help of the brake examiner) per man. . .	20
That is for dealing with the whole train from an examination point of view.	60

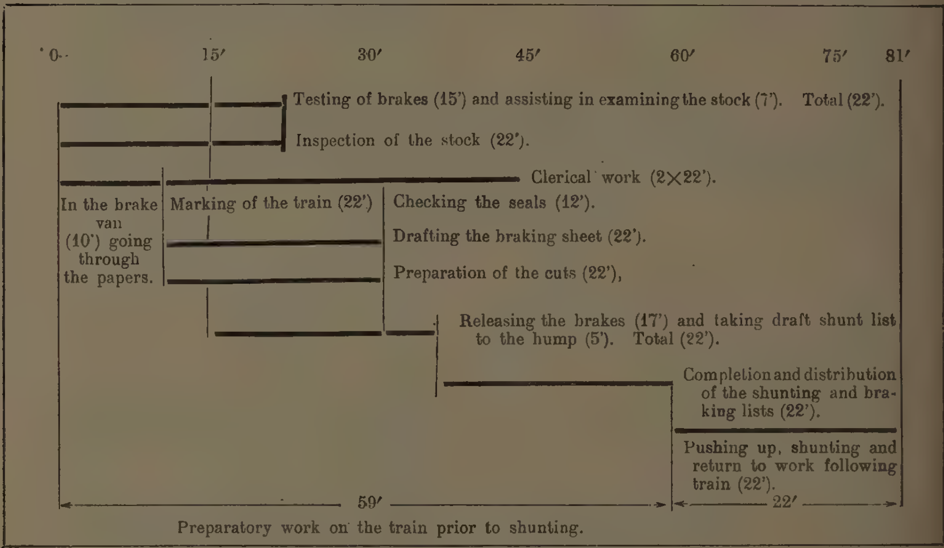
3. *Preparatory work for shunting.*

a) Foreman shunter :	
Drawing up part of the shunting list (for copy see appendix 3) according to the marking of the wagons as done by the clerk mentioned previously	20
b) 1st shunter :	
Preparation of the cuts from the indications given by the foreman shunter	20
c) 2nd shunter :	
Releasing the brakes and uncoupling the train pipe at the end of each cut (15 min.) and taking the draft shunt list to the hump and returning (5 min.). . . .	20
d) Completion of the shunting list by the assistant in charge of the hump (copy given in appendix 3) ; copying the brake list (for copy see appendix 4), taken by a junior clerk and distribution to the pointsmen and brakesmen	20
That is for completely handling a train as regards preparatory work before shunting (1). . .	80

(1) As will be seen later on, this time of 80 minutes is reduced if the work of these men partly overlaps.

B. — Shunting with a single shunting engine.

	Minutes.
Pushing the rake up to the hump top	5
Shunting the 60-wagon train . .	12
Running back the locomotive. Allocation to another rake . . .	5
so that the cadence of shunting is	22



Note. — In this table sign ' = Minute.

* * *

The operations involved in getting the train ready for shunting take place to some extent simultaneously as shown in the above table.

The shunting operations follow on immediately.

* * *

Preparatory work on the belt system.

The application of the belt system is based on the same time being required for each of the operations to be effected according to the desired rhythm of shunting.

It should be noted, however, that good organisation implies continuity of work, which should also be distributed uniformly over each turn of duty. If the number of trains arriving during one turn is small, the period between two successive push-ups can be increased in order to reduce the number of men or shunting engines used. It would appear to be possible also by momentarily accelerating the output to include in the work

done in a turn, an occasional special train.

At Antwerp-North during the periods 6 a.m. to 2 p.m., and 2 p.m. to 10 p.m., only 15 and 13 trains come in respectively, whereas at night, between 10 p.m. and 6 a. m., 22 trains have to be handled (20 booked and 2 as required).

The phase duration during these periods has to vary in consequence for reasons of economy :

It will be 17 minutes in the 10 p.m. to 6 a. m. turn, 22 minutes in the 2 p.m. to 10 p.m. turn, and 40 minutes in the 6 a.m. to 2 p.m. turn of duty, as described below.

6 a. m. to 2 p. m. period.

Between 6 a.m. and 2 p.m., the number of trains received makes it impossible to keep the single shunting engine used continuously at work.

The cadence of the shunting which could be 22 minutes cannot therefore determine the duration of the working phase unless the staff is to stop work at certain moments. In order to ensure continuity of operations, and having regard to the fact that the wagons received are generally for the port where they can only be put into position as a rule between 5 p.m. and 8 a.m., it is possible, for the sake of economy, to slow down the work done in the 6 a.m. to 2 p.m. period so as to bring into the 2 p.m. to 10 p.m. period the number of trains needed to organise the work to a cadence of 22 minutes.

The phase during the 6 a.m. to 2 p.m. period will then depend on the number of trains to be shunted in order to leave a carry-over sufficient to ensure continuity of work between 2 p. m. and 10 p.m. at the 22-minute cadence.

Losses of time through drawing out the trains, and through having to back work wagons [1], result in it being possible to shunt 18 to 19 trains at this 22-minute cadence during the 2 p.m. to 10 p.m. period; it is necessary during the 6 a.m. to 2 p. m. period to get ready for shunting 11 of the trains received which gives rise to a phase of *40 minutes* [2].

In order to equalise the times for each operation, the clerical work must be covered by two men, thereby bringing the time to $\frac{40 + 30}{2} = 35$ minutes or 40 minutes with 10 % margin.

The preparation of the brake list, taking it to the hump, releasing the brakes and getting ready for the cuts takes a foreman shunter 40 minutes; part of these operations will be covered by the shunter at the sidings from which trains leave for Zurenborg.

Testing the brakes and examining a train requires 66 minutes' work; two examiners will be sufficient to cover it in the phase of 40 minutes. The loss of time will disappear when the yard will be fitted with the necessary installation for testing the brakes.

As regards completing the shunting list

(1) Back working is the operation of clearing the head of the fan of the shunting sidings, at the hump end, of wagons which for various reasons have stopped there too soon. by collecting them together on each line by means of a shunting engine or in certain cases by pinch-bars.

(2) At the beginning of the turn of duty the time lost as a result of the trains being spaced more than 40 minutes apart amounts in all to 40 minutes, leaving 8 hours — 40 minutes = 440 minutes for the preparatory work on 11 trains, *i. e.* 40 minutes per train.

and drawing up and distributing the brake sheet, the slowing down of operations enables this to be covered by the assistant foreman at the hump. This organisation will *economise four men* (two clerks and two shunters) as compared with the position prior to the introduction of the belt system.

2 p. m. to 10 p. m. period.

In view of the arrangements made for the 6 a. m. to 2 p. m. period, the phase duration can be fixed at 22 minutes. In order to equalise the time spent on each operation, the clerical work must be covered by two men, thereby reducing the period to $\frac{40}{2} = 20$ or 22 minutes with 10 % margin.

The office work, 30 minutes' time, is covered by a man called « chargeman » (chef de poste). As the time of 30 minutes exceeds the phase of 22 minutes by 8 minutes, the office work has to be accelerated by 8 minutes; this has been done by making the telephonist assist in the work or, if need be, the two men on outside duty.

The foremen-shunters, shunters, and examiners, have also been allowed a 10 % margin.

If we suppose that, at a given moment, the reception sidings are completely free and that the trains come in at regular intervals of 22 minutes, each one will be pushed up the hump at the 59th minute after its arrival (see page 210).

Moreover, if the trains follow one another at intervals of less than 22 minutes the organisation of the work on the belt system will ensure the shunting being done at regular intervals of 22 minutes.

10 p. m. to 6 a. m. period.

The marshalling yard during this period should deal with the arrival of 22 trains (20 booked and 2 as required) and despatch 10 trains from C_2 to C_1.

The time to despatch a train from C_2 towards C_1 is made up of :

3 minutes for attaching the engine at the head of the train;

7 minutes for the train to get away and clear the points, that is, for the 10 trains $10 \times 10 = 100$ minutes.

All these trains do not cause interruptions of this length, it being possible to send the trains from sidings 2 to 5 without interrupting the shunting, but a certain percentage must be allowed for the inevitable back working of wagons, so that for the two causes of delay : *a*) despatching the trains, *b*) back working wagons, a total loss of 100 minutes in eight hours must be allowed.

To make good this loss and deal with a rapid succession of incoming trains, two locomotives must be used for hump shunting, which in theory reduces the shunting time by 5 minutes + 5 minutes = 10 minutes (in practice the gain is only 5 minutes).

The phase period should therefore be reduced to 17 minutes [1].

To get equal intervals of time the work should be regulated as follwos :

Clerical work : outside work at the train : 3 men each working 17 minutes $\left(\frac{40}{3}\right.$ + working margin).

Office work :

2 men, wich gives $\frac{30}{2} = 15$ minutes, or with rather more than 10 % margin ;

(1) 8 hours — 100 minutes = 380 minutes. 380 minutes : 22 trains = 17 minutes.

17 minutes (chargeman assisted by telephone operator).

Work of the leading shunter and shunter :

1st gang : 1 leading shunter and 1 shunter.

2nd gang : 1 leading shunter and 1 shunter, $\frac{60}{4} = 15$ minutes, that is, with rather more than 10 % margin, 17 minutes.

Brake examiner : average time, 15 minutes, with slightly more than 10 % margin : 17 minutes.

Rolling stock examiner : 3 examiners $\frac{60}{3} = 20$ minutes, are assisted by the brake examiner.

Development of the harmonograph.

The harmonised work as it is done at the present moment has been translated into a harmonograph, an extract from which is reproduced in appendix 5.

This diagram gives, as a function of the time, the graphical representation of the succession of operations effected by each man.

The work done by a man being shown by rectangular bands marked in the same way, we see that the occupations of each are continuous and that they are so linked together that the cadence of shunting be 22 minutes.

The dotted line represents the light working of the shunting engine to get behind the rakes to be shunted, whilst the oblique line indicates the pushing over the hump, and the heavy horizontal lines, where marked « hump », show the time taken for each rake to go over the hump.

***

In order to ensure the work being properly done within the given time, work cards (see appendix 6) show each man for each turn of duty, the cycle of operations to be done during his turn of duty.

The drawing up of a harmonograph is not sufficient in itself : it must be followed up carefully, experience showing the changes to be made to it. For this reason each chargeman has to make a brief report in a prescribed form on the way the service has been carried out. Any untoward happenings are briefly described, so that after enquiry the harmonograph may be suitably modified.

Naturally the working conditions may be altered considerably by wind, frost, late arrival of trains, accidents, etc. The harmonograph none the less represents a grouping together of the elements and so constitutes a sure guide for the whole of the operations.

Operations in fan C_2.

These subdivide themselves into :

1. Braking the consecutive wagons on shunting;

2. Formation of the trains for despatch to the port or to Holland.

These operations should be organised in harmony with the belt method of working which has been described, so that braking can be done perfectly at the selected rate of shunting and that the formation of the trains in geographical order may be completed within the desired time so that the C_2 fan of sidings may be cleared in rhythm with that of the reception of trains to be broken up.

1. *Braking the wagons.*

The wagon braking is covered by two lines of brakesmen; these men are distributed so as to get two in the front line and 8 beyond.

The braking sheet makes it possible to get full use out of the men in the second line as a result of a distribution of work whereby some of them have a sufficiently long interval between two brakings for them to travel with the cuts and use the hand brake.

During the period the shunting is stopped, the wagons are buffered up with pinch-bars and coupled together. In some cases the wagons are buffered up by one of the locomotives happening to be at the outgoing end of the C_2 yard.

2. *Formation of the trains or rakes prior to departure.*

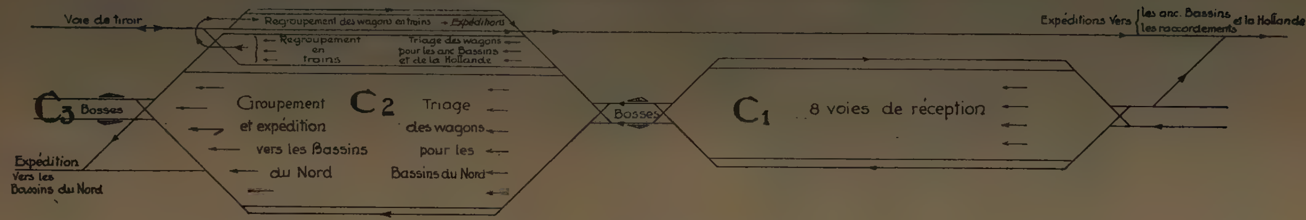
A record is kept of the occupation of the sidings in group C_2. After each shunt, and by using the information of the shunting sheet, the assistant station master responsible for sending away the trains and rakes is advised of the number of vehicles sent on to each siding of the C_2 group and regulates the despatch of the rakes accordingly.

Appendix No. 7 reproduces part of the record taken. Each track of group C_2 is shown by two columns. In one is given the total number of wagons on the siding after each shunt, and in the other the number of vehicles despatched.

The two first columns are used to enter the number of the train or rake and the time of arrival or departure.

Working at the slower pace which makes possible a rational distribution of the work and greater regularity in carrying it out, has as a consequence the reduction in the number of individuals in the different grades of staff who take part in preparing trains for shunting.

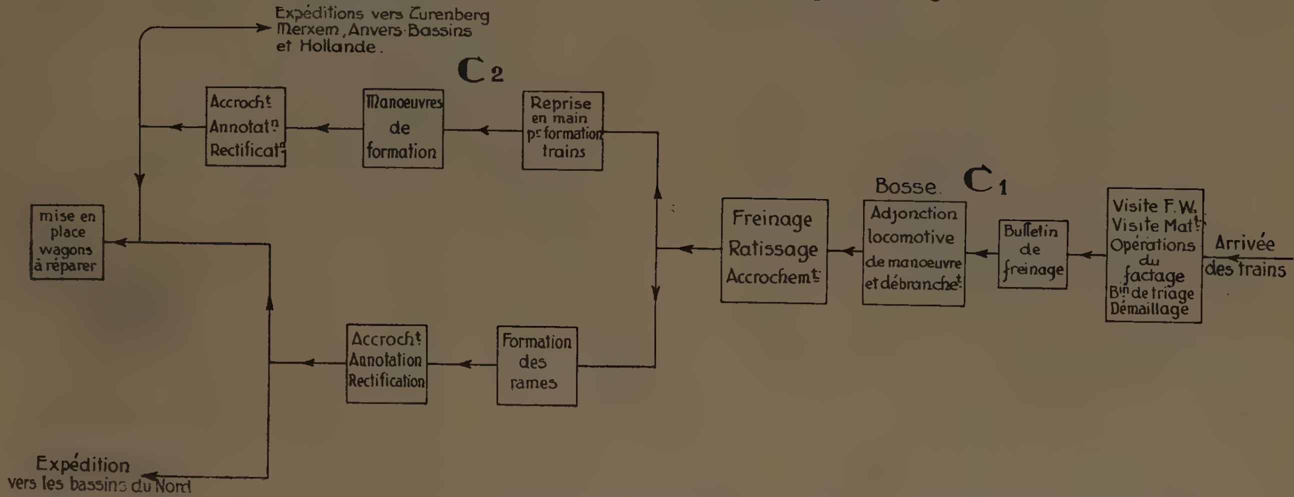
Diagram of movements from the train arrival at C_1 until departure from C_2.



Appendix 2.

Antwerp-North.

Diagram of the successive operations the wagons undergo.



Explanation of French terms in appendices 1 and 2:

Accrochement, annotation, rectification = Coupling up, recording, correcting. — **Adjonction locomotive de manœuvre et débranchement** = Attaching locomotive and shunting. — **Arrivée des trains** = Train arrival. — **Bosses** = Humps. — **Bulletin de freinage** = Brake list. — **Expédition vers les anc. Bassins et la Hollande (les raccordements)** = Despatches for the old docks and Holland (the junctions). — **Expédition vers les Bassins du Nord** = Despatch to North Docks. — **Formation des rames** = Formation of rakes. — **Freinage, ratissage, accrochement** = Braking, buffering up, coupling. — **Groupement et expédition vers les Bassins du Nord** = Grouping and despatch towards the North Docks. — **Manœuvres de formation** = Marshalling. — **Mise en place wagons à réparer** = Placing wagons for repairs. — **Regroupement en trains** = Forming into trains. — **Regroupement des wagons en trains** = Forming wagons into trains. — **Reprise en main pour formation trains** = Collecting wagons to form trains. — **Triage des wagons pour les bassins du Nord** = Shunting the wagons for the North Docks. — **Visite FW. Visite mat. Opérations du factage. Bulletin de triage. Démaillage** = Brake examination. Stock inspection. Clerical work. Shunt list. Cuts. — **Voie de tiroir** = Draw-out track. — **8 voies de réception** = 8 reception sidings.

Antwerp-North Yard.

Appendix 3.

Shunting list.

Date : . . . . Train No. . . . . . at . . . . h. . . m. . . . , siding . . . . . .

Order number of shunt.	Number of the siding.	Number of wagons.	Total gross weight.	Cuts. First wagon. Gross weight		Brake percentage of each cut.	Braking length. Brakesmen. 1st line		Second line.								Remarks.
				Gross weight	Number of axles		a	b	1	2	3	4	5	6	7	8	
To be filled in by foreman shunter.							To be filled in by assistant in charge of the hump.										

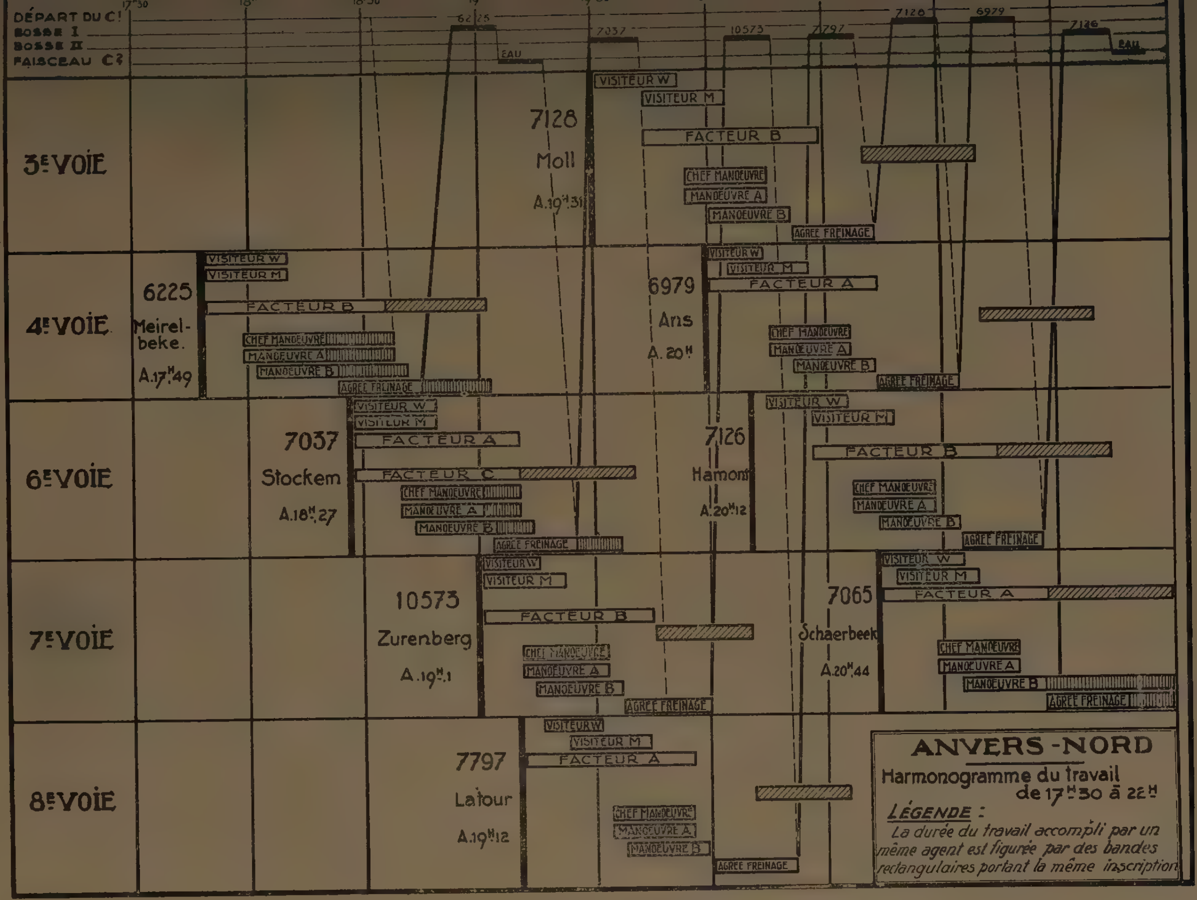
Appendix 4.

Brake list.

Train . . . . from . . . .

. . . . siding, arrived at . . . . . . h. . . . . m.

Number.	Siding.	Wagons.	Braking length.	Remarks.



Explanation of French terms:

Agréé freinage = Clerk drawing up braking lists. — Bosse = Hump. — Chef manœuvre = Foreman shunter. — Départ du C¹ = Departure from fan of sidings C¹. — Facteur = Clerk. — Faisceau = Fan of sidings. — Harmonogramme du travail.. = Harmonograph of the work between 5.30 p. m. and 10.0 p. m. — La durée du travail accompli... = The duration of the work accomplished by one given man is shown by the rectangular bands marked in the same way. — Manœuvre = Shunter. — Visiteur = Examiner. — Voie = Track.

Appendix 6.

Work sheet to be completed for each service by the clerks.

Date . . . .

Clerk *a*, *b* or *c*. — Service I, II or III.

Succession of trains.	Arrival siding.	Work in the reception sidings.		Office work.		Remarks.
		started.	ended.	started.	ended.	

Signed : . . . .
Sign-manual of chargeman : . . . .
Sign-manual of head clerk : . . . .

Appendix 7.

Antwerp-North (marshalling yard).

Date . . . .

Trains.		Rakes.			Shunting siding.							
Arrival time.	Number.	Arrival time.	Destination.	Number of shunting engine.	2		3		4		etc.	
					A	D	A	D	A	D	A	D

[625. 143.5]

The life of rails,

by Dr. Ing. P. MAST, Consulting Engineer, Breslau.

(*Zeitung des Vereines deutscher Ingenieure*, vol. 74, No. 48.)

Analytical investigation of the abrasive forces exercised on the rails by the locomotive and train and their effects on the wear thereof on level, straight and inclined sections, on lengths where there is braking, and on curves. Example for the approximate calculation of the life of the rails of a given section of line.

The data given in technical literature and the statistics on the life of rails, are so contradictory that neither economical and financial decisions nor dependable calculations of the maintenance costs can be based on them. The contradictions are due to the fact that the statistics up to the present have been compiled without previous classification of the *causes of the wear of the permanent way*. The large variations in the wear of the rails recorded, cannot be explained by differences in the quality of the steel alone.

Facts regarding the wear of rails, as known up to the present.

The statistical results given by the Associated German Railway Companies (*Verein*) for the year 1912 are shown in table I.

The following facts were taken from various publications at the disposal of the author :

A. M. Wellington (1), the American veteran investigator of railway economics, gave as the useful life of rails weighing 27 to 36 kgr. (54.4 to 72.6 lb. per yard) a value of 135 to 180 million tons or 300 000 to 500 000 trains. The total wear of the rails amounts to 5 to 7.5 kgr. (10 to 15 lb. per yard) whereof the wear of 0.45 kgr. (0.9 lb. per yard) corresponds to a gross load of 9 million tons, or 10 to 16 mm. (0.394 to 0.630 inch) of the head of the rail, the wear of 1.6 mm. (0.063 inch) corresponding to a gross load of 12.7 to 13.6 million tons. These figures are admittedly not of recent date and do not take into account any subsequent improvements which may have been made in the quality of the steel.

Webb, another American author, gives the results of trials on the Northern Pacific Railroad on two straight inclined sections with gradients of 1 in 333 and 1 in 191 and on a level section at the lower end of an up-grade all almost equally stressed (2).

With a traffic of 9 million tons, the loss of weight of the rails amounted to 0.648 % of the eight of the rail on the 1 in 333 gradient, 0.401 % on the 1 in 191 gradient, and 0.260 % on the level.

If the latter value is applied to a rail with a number 8 section, the maximum wear of which is 8 % of the total weight of the rail, the life of the rail on the level would amount to $\frac{8}{0.26} \cdot 9 = 276$ million tons service load per track (not per rail).

In his criticism of the trial results communicated by the Northern Pacific,

(1) A. M. WELLINGTON, *Railway Location*, 1906, page 119.

(2) WEBB, *The Economics of Railroad Constructions*, 1st edition, p. 157.

TABLE I.

Summary of the results of the rail statistics of the « Verein » for the year 1912 [1].

Number of trial sections	Number of rails measured	TOTAL RESULT.	To a reduction of 1 mm. of the height corresponds a maximum gross load, in millions of tons, of	To a reduction of 1 mm² of the surface corresponds a maximum gross load, in millions of tons, of	To a reduction of 1 mm. of the height corresponds a minimum gross load, in millions of tons, of	To a reduction of 1 mm² of the surface corresponds a minimum gross load, in millions of tons, of	To a reduction of 1 mm. of the height corresponds an average gross load, in millions of tons, of	To a reduction of 1 mm² of the surface corresponds an average gross load, in millions of tons, of
		Single-line test lengths :						
23	160	Martin steel	41	0.6	10	0.2	19	0.3
23	148	Thomas steel	58	2.1	19	0.4	37	1.2
86	623	Bessemer steel	205	4.5	7	0.1	38	0.8
132	931	Single-line test lengths	205	4.5	7	0.1	34	0.8
		Maximum and minimum value of statistics for 1909	111	1.1	6	0.2	25	0.8
		Maximum and minimum value of statistics for 1906	884		5			
		Double-line test lengths :						
178	1011	Martin steel	259	4.1	9	0.1	55	1.1
199	1490	Thomas steel	15	2.3	12	0.1	36	0.7
70	607	Bessemer steel	125	3.3	19	0.3	53	1.1
447	3108	Double-line test lengths	259	4.1	9	0.1	46	0.9
		Maximum and minimum value of statistics for 1909	150	3.4	10	0.2	38	0.9
		Maximum and minimum value of statistics for 1906	271		4			
		All test lengths :						
201	1171	Martin steel	259	4.1	9	0.1	49.1	1
222	1638	Thomas steel	152	2.3	12	0.1	42	0.7
156	1230	Bessemer steel	205	4.5	7	0.1	43.8	0.9
579	4039	All test lengths	259	4.1	7	0.1	42.8	0.9
		— — — in 1909	150	5.2	6	0.2	33.4	0.8

(1) Röll, *Enzyklopädie des Eisenbahnwesens*, Berlin, 1917, 2nd edition, vol. 8, p. 316.

Webb also mentioned the law found by *Stockert*, that the wear of the rails is less during the first than in the later years of service.

The same author furthermore gave several figures of the wear of rails in curves measured in degrees (3), the average results of which, converted

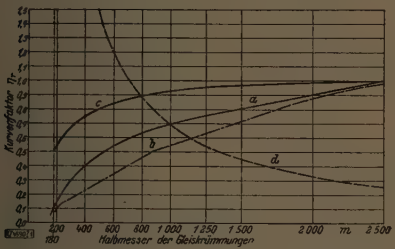


Fig. 1. — Curve factor η_{r} and curve resistance w_{r} plotted against the radius of curvature.

a) According to Webb;
b) According to Pennsylvania Railroad;
c) Theoretical, based on calculation of curve resistance;
d) Graph of curve resistances according to Röckl

$$w_{r} = \frac{650}{R - 55} \text{ kgr. per ton.}$$

Note. — Kurvenfaktor = Curve factor. — Halbmesser der Gleiskrümmungen = Radius of curves.

into metric units, are plotted in figure 1, curve *a*.

The ordinates of the curve factor graph η_{r} represent the fraction by which the life of the rail on the level must be multiplied to give the life on the curve.

A chief engineer of the Pennsylvania Railroad gave, by the Webb method, the values shown in table II as the average life of rails in curves on the main line of this company.

The curve factor corresponding to the data of the Pennsylvania Railroad, is shown by graph *b*. In addition a theoretical graph *c has* been plotted showing the curve factor which was calculated on the assumption that the rails are worn in proportion to the forces acting, that the average value of the force acting on the rail due to friction and rolling resistance on the straight level may be taken as 5 kgr. (11 lb.) per ton gross load, and that this force is increased by the curve resistance (graph *d*).

A comparison of the graphs *a* to *c* shows that the grinding action of the frictional forces occurring in the curves on account of lateral pressure, acceleration and retardation of the wheels (as well as the friction due to the braking forces) is far more destructive than the rolling resistance of the vehicles and is probably also greater than the wear caused by the frictional forces of the locomotives.

The American investigations also refer to the characteristics of the inside and outside rails on curves, where they, as well as the German Railways (*Verein*), have found more wear on the outside rail, the wear gradually becoming more evenly distributed on curves of larger radius.

Proposed new investigations into the wear of rails. The abrasive forces.

The problem of the wear of rails has therefore only been partly solved; closer statistical observation of the available *service sections* is required, because tests must not be exclusively confined to trial sections.

The fatigue and wear of the rails, apart from the influence of damp, frost, etc, is caused by:

1. the effect due to the load of the cars and locomotives, which consists of rolling forces acting in a forward direction (rolling resistances);
2. the effect due to lateral pressure of the cars and locomotives caused by guiding forces and wind pressure on the straight, manifesting itself in alternate up and down movements; forces acting alternately in a forward and rearward direction, due to une-

(3) In America the curves are specified by the angle which an arc of 100 feet subtends at the centre.

TABLE II.

Average life of curve rails on the Pennsylvania Railroad.

Degrees for arc measuring 100 feet in length.	Level.	1°	2°	4°	6°	8°	9°
Radius of curvature. *metres* . . . / *chains* . . .	∞	1746.5 (*87.2*)	873 (*43.6*)	436.5 (*21.8*)	294 (*14.7*)	218 (*10.9*)	194 (*9.7*)
Life in months	120	96	60	30	20	14	9
Curve resistance according to Röckl, referred to the weight of the train, kgr./t. (*lb. per Engl. ton*) . . .	0	0.36 (*0.80*)	0.80 (*1.8*)	1.70 (*3.8*)	2.76 (*6.2*)	3.99 (*8.95*)	4.68 (*10.5*)

ven rolling surfaces of the tyres, giving rive to a to and fro sliding movement on the rolling surface of the rails and accompanying the nosing movements of the vehicles or the action of a side wind;

3. the vertical and slanting blows of the vehicles on the rail caused by uneveness of the rails, especially at joints and by wheel flats;
4. the forces acting in curves partly in a forward and partly in a rearward direction caused by the retardation and acceleration of the wheels, and aslo by the resistance to the increased guiding forces in curves (curve resistances);
5. the driving force exerted by the locomotive on the permanent way, acting rearwards and having a sliding effect;
6. the braking forces exerted by the cars and the locomotive on the permanent way, acting forwards and having a sliding effect;
7. the reactions to the forces 1 to 6 which, contrary to these, do not act on the head of the rail, but on the foot. Exceptions are the reactions acting on the rail head and on the foot through the fish plates.

It appears from this list that some of the forces have a double effect as they act directly on the rail and also influence friction. If the reactions are also taken into consideration, a threefold effect of the forces acting directly on the rail is sometimes found.

Apart from the more important causes of wear enumerated above, a few others still remain to be mentioned. The influence of the speed of the trains is fully taken into account by calculating the resistances by the formula $(a + b\,v^2)$. *Ehrensberger* calls attention to statements in the technical press [4] according to which the wear of the rails is proportional to the square root of the wheel pressure. The accuracy or otherwise of this law is a question to be settled by laboratory experiments.

The influence of local climatic conditions other than wind effects on the rolling can be ascertained by laying down alongside the service rail under test, a new rail, the loss of weight of which is measured at the same time. The influence of wind on the moving train must be tested by ascertaining the direction of the trial rail with respect to that of the prevailing winds. The

[4] *Organ für die Fortschritte des Eisenbahnwesens*, vol. 80 (1925), p. 303.

trials made by *Strahl* show that this influence should not be underrated.

In order to separate the effects of the reactive forces mentioned under 7 from those under 1 to 6, the wear of the rail must be measured by instruments provided *ad hoc* in addition to ascertaining the total loss of weight. The wear of the rails due to the reactive forces occurs at parts of the rail section different from those caused by the direct forces. The influence of the reactive forces can therefore be determined with fair certainty.

A very important question, which is yet to be solved, bears on the relation between the wear of the rails due to rolling resistance and that due to sliding resistances. The answer can only be found by laboratory tests, or at least be suggested thereby, if it is to be satisfactory. If the number of test rails in the same section, all carrying an exactly equal amount of traffic, is large enough, it is probable that the influences of the forces enumerated under 1 to 6 can be determined by means of the theory of probability and by equations involving several unknowns.

Those lines over which only trains of the same composition and carrying equal loads run, e. g. trains used for conveying ore, coal, or other goods in bulk, are naturally most suitable for observation.

For each trial the quality of the steel of the rails and their service age must be given, because both factors influence the results of the test to a considerable extent. It is specially important to determine exactly the forces which are likely to produce the greatest effects, *i. e.* the rolling resistance of the train, the tractive power of the locomotive, the curve resistances, and the brake forces. The pull is best measured by means of a recording tension dynamometer, as described in the test made by *Strahl* [5].

[5] STRAHL, *Zeitschrift des V.D.I.*, vol. 57 (1913), p. 328.

A tentative method of calculation is given at the end, by which the life of the rails of new lines can be determined by estimating their probable wear.

Development of several formulæ for the determination of the abrasive forces.

The following abbreviations will be used :

- G Weight of train including locomotive, in metric tons,
- L Total weight of locomotive including tender, in metric tons,
- L_1 Weight on driving axles of locomotive, in metric tons,
- L_2 Weight on carrying axles of locomotive and tender, in metric tons,
- Q Weight of the cars, in metric tons,
- w_w Frictional resistance of the cars, in kgr. per metric ton,
- w_L Frictional resistance of the locomotive in kgr. per metric ton,
- w_r Curve resistance in kgr. per metric ton,
- n Number of driving axles,
- V Speed of train in km. per hour.

When the *vehicles* are running with the *brakes off*, a rolling resistance results, which can be assumed to be equal in amount on the level and on inclines. In addition to this, there are guiding and wind resistances which can also be taken as being equal on all straight sections of a line, as long as they are in the same relative position to the direction of the prevailing wind.

The train resistance formula taking all frictional resistances of the cars into consideration is :

$$w_w = 2 + (0.007 + m)\left(\frac{V}{10}\right)^2 \pm s + w_r \text{ kgr. per metric ton hauled} \quad (1)$$

In this formula the value 2 comprises the resistance due to the wave deflection of the rail, the elastic flattening of the wheel, the guiding resistance, the shock resistance due to wheel flats, certain retarding and accelerating forces result-

ing from unequal tyres, and axle friction.

The value $0.007\left(\frac{V}{10}\right)^2$ is the resistance due to the joints in the rails, and $m\left(\frac{V}{10}\right)^2$ that due to air resistance.

These resistances act directly on the rails with the exception of axle friction, air resistance and resistance due to gradients.

The axle friction, according to *Hütte,* 25th edition, Vol. 2, due to semi-fluid friction and air-cooling amounts to about 0.6 kgr. per ton hauled [6].

The resistances directly set up by the cars on the rails, mentioned above under 1 to 3, therefore amount to :

$$w_w = (2 - 0.6) + 0.007\left(\frac{V}{10}\right)^2 = 1.4 + 0.007\left(\frac{V}{10}\right)^2 \text{ kgr. per ton.} \quad (1a)$$

The side wind pressure is considered in the formula (1) when the value $m\left(\frac{V}{10}\right)^2$ is increased to $m\left(\frac{V+12}{10}\right)^2$ for a moderate wind, and to $m\left(\frac{V+18}{10}\right)^2$ for a strong wind. The frictional forces caused by a moderate side wind, acting directly on the rail amount to :

$$m\left[\left(\frac{V+12}{10}\right)^2 - \left(\frac{V}{10}\right)^2\right] = m(1.44 + 0.24\,V) \text{ kgr. per ton.}$$

The resistance calculated by Strahl for the *locomotive* itself [7], when there is no side wind, is :

$$w_L = cL_1 + 2.5L_2 + 0.6F\left(\frac{V}{10}\right)^2 + 0.004Z_t \text{ in kgr. per ton.}$$

For large locomotives $F = 10$ m², $c = 5.8$ to 9.5 according to the number of the driving axles and cylinders, and Z_t is equal to the indicated tractive effort. The value c takes account of the rolling resistance and the influence of the driving mechanism and can be replaced by the expression $\sqrt[5]{(2.3\,n)^4}$. The constant 2 in the car formula corresponds to the constant 2.5 in the locomotive formula. The term $0.007\left(\frac{V^2}{100}\right)$ representing the resistance due to the joints in the rails in the case of the cars, should be increased for locomotives in the ratio $\frac{2.5}{2.0}$, i. e. to $0.009\,\frac{V^2}{100}$.

The *Strahl* formula for locomotives with 2 and 4 cylinders can therefore be written as follows, separating the various forces :

$$w_L = \left[2.4 + 0.009\left(\frac{V}{10}\right)^2\right](L_1 + L_2) + \sqrt[5]{(2.3n)^4}\,L_1 + 0.6F\left(\frac{V}{10}\right)^2 + 0.004Z_t + w_r \pm s \quad (2)$$

Subtracting the axle friction, the frictional resistances 1 to 3 which, besides the frictional resistance of tractive effort of the locomotive, act directly on the rails are :

$$w_L = \left[1.8 + 0.009\left(\frac{V}{10}\right)^2\right](L_1 + L_2 \text{ kgr. per ton} \quad (2a)$$

The influence of lateral wind of moderate strength according to the *Strahl* formula is :

$$0.6F\left[\left(\frac{V+12}{10}\right)^2 - \left(\frac{V}{10}\right)^2\right] = 0.6F(1.44 + 0.24V).$$

For calculating the *frictional force* Z_e on the driving wheels, the formulæ (1) and (2) must be used provided the required additions for lateral wind are made.

The curve resistances are undoubtedly larger at the locomotive wheels than at the car wheels. According to *Röckl* the

(6) The axle friction, at starting, is a multiple of this value.

(7) STRAHL, *Zeitschrift des V.D.I.*, vol. 57 (1913), p. 251.

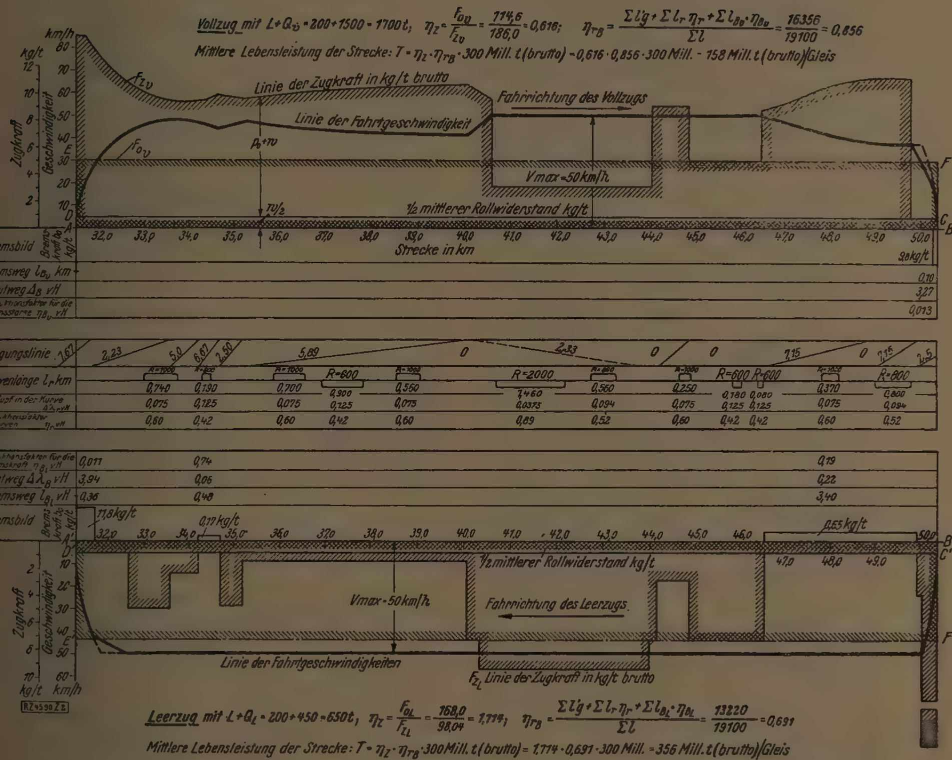


Fig. 2. — Tractive force and resistance diagrams for full and empty trains of an industrial haulage railway, referred to the gross weight of the train in kgr. per ton. — Example for the calculation of the average life of a railway line in tons (gross).

Explanation of German terms:

Bremsbild = Braking diagram. — Bremskraft = Braking force. — Bremsweg = Braked length. — Fahrrichtung des Vollzugs = Running direction of train fully loaded. — Geschwindigkeit = Speed. — Gleitweg = Sliding length. — Kürvenlange = Length of curves. — Leerzug mit... = Empty train with... — Linie der Fahrtgeschwindigkeit = Graph of running speeds. — Linie der Zugkraft *kg/t* (brutto) = Graph of tractive efforts in kgr. per ton (gross). — Mittlere Lebensleistung der Strecke = Mean life of track section. — Mill. *t* (brutto) Gleis = Millions of tons (gross) track. — Mittlerer Rollwiderstand = Mean rolling resistance. — Neigungslinie = Gradient line. — Reduktionsfaktor für die Bremsstärke = Reduction factor for the brake force. — Reduktionsfaktor in Kurven = Reduction factor in curves. — Schluf in der Kurve = Slip in curve. — Strecke in *km* = Distance in kilometres. — v H = Per cent. — Vollzug mit... = Fully loaded train with... — Zugkraft = Tractive effort.

curve resistance of the complete train including the locomotive is :

$$w_r = \frac{650}{R-55} \text{ kgr. per ton,}$$

where R is equal to the radius of the curve in metres. The curve resistance of the cars according to *Hütte*, vol. 3, 25th edition, p. 797, amounted to $\frac{520}{R-55}$ kgr. per ton.

Therefore the resistance of the train works out at :

$$(G - L)\, w_{r_w} = (G - L)\frac{520}{R-55} \text{ kgr.}$$

The curve resistance of the locomotive is consequently :

$$w_{r_L} = \frac{650}{R-55}\left(1 + \frac{Q}{5L}\right).$$

This formula for the curve resistance of the locomotive is not sufficiently satisfactory and should be replaced by another not based upon Q, after trials have been made, and at the same time taking into account the investigations conducted by Jahn [8].

A comparison of the records taken of the section will show to what extent the trials must take account of the maximum or mean rigid wheel base, the wheel arrangement of the locomotive and vehicles, the speed of the train, the ratio between the weight of the locomotive and that of the vehicles, etc. The full calculated value of the curve resistance of the vehicles acts directly on the rails, and also indirectly by increasing the frictional forces produced by the locomotive.

(8) It should be mentioned that the last formula gives very high curve resistances for the locomotive. The tractive effort of the latter is accordingly reduced on curves to a far greater extent than has been assumed up to the present, and when designing new stations, care should be taken that when starting locomotives do not stand on curves.

Exemple of a tentative calculation to determine the life of the rails.

In order to determine the life of the rails of a newly planned section of fair length, I have adopted a new method, involving a graphical representation of the pull and the resistances, referred to 1 ton gross weight, for the kind of trains running on the particular line. The initial value for the life of the S 49 rails when used on a *level straight* line, was assumed as 300 million tons gross weight for the track.

First, the *rolling resistances* of the cars and of the locomotive of the train, Qw_w and Lw_L are determined in kgr., according to the formulæ (*1a*) and (*2a*) for both directions of traffic, with various conditions of loading and for a full and empty train. The mean rolling resistance when V = 30 km. per hour is then :

$$w = \frac{1.49\,Q + 1.91\,L}{Q - L} \text{ kgr. per ton.}$$

The rolling resistance was assumed to be half the frictional resistances of the locomotive, and is represented by $w/2$ plotted above the lines *AB and A'B'* of figure 2.

The tractive forces. — The graph showing the tractive efforts of the locomotive per ton weight of the train is obtained at the same time that the graph is being made by means of the *Unrein* method [9]. It is well known that the gradient depicted in figure 3, which the locomotive can work over at the given speed is equal to the acceleration force per ton weight of train with the regulator open on the level, $p_0 = \frac{Z - W}{L + Q}$, where Z is the tractive force of the locomotive and W is the total resistance of the train.

(9) *Organ für die Fortschritte des Eisenbahnwesens*, vol. 79 (1924), p. 117.

Further, on an up gradient s v T.:

$$p_s = \frac{Z - W - (L + Q)s}{L + Q} = p_0 - s \text{ kgr. per ton,}$$

and on a down gradient — s v T. :

$$p_{-s} = \frac{Z - W + (L + Q)s}{L + Q} = p_0 + s \text{ kgr. per ton.}$$

If $W = w\ (L + Q)$,

$$p_0 = \frac{Z}{L + Q} - w;$$

$$p_s = p_0 - s = \frac{Z}{L + Q} - w - s;$$

and

$$p_{-s} = p_0 + s = \frac{Z}{L + Q} - w + s.$$

The corresponding tractive forces of the locomotive with the regulator open per ton weight of train work out in all three cases, from the above formulæ, to :

$$\frac{Z}{L + Q} = p_0 + w \quad . \quad . \quad . \quad . \quad \text{(I).}$$

When the regulator valve is closed $Z = 0$ and the resulting retarding forces are as follows : on the level $p'_0 = -w$; on a rising gradient $p'_s = -(w + s)$ and on a down gradient $p'_s = s - w$.

The s/V and w/V lines shown in figure 3 hold good for a works locomotive of 120 tons adhesive weight, 1 700 tons gross train weight and 650 tons empty train weight.

According to formula 1 and provided the maximum speed V is not reached. the tractive forces holding good for the separate points on the line $(p_0 + w)$ [or $(s + w)$ — see below] can be taken from the graph, if besides the s/V line the w/V line is also plotted below the axis of the abscissæ. The tractive forces $(p_0 + w)$ are then plotted in figure 2 above the graph of the rolling resistances in order to obtain the graph of the tractive effort per ton weight of train. The lines EF and E′F′ show the graphs of the tractive effort for the full and empty train over an equal length of straight and level track.

If according to figure 3, the accelerating force corresponding to the maximum speed is designated by p_V, the corresponding resistance of the train itself being equal to w the limiting gradient on which the train is able to run without losing speed is equal to $s_V = p_V$.

Let this be called « the ruling speed gradient » and the limiting gradient designated so far as the ruling gradient, « the ruling friction gradient ». The gradient down which the train moves with the regulator *closed* without loss of speed, is $s_B = -w$.

In the case of the maximum speed V, $s_B = -w_V$. All down gradients which are steeper than w_V should be described as brake gradients, if the train is already running at the maximum speed when coming onto them.

The run with the regulator closed and the two further cases with partly open regulator and brake applications must be specially considered. The locomotive drivers operate the regulator by ear as soon as they have reached the maximum speed, as they find the periodic jar due to the rail joints sufficient to serve as a maximum speed indicator. The speed indicator of the locomotive is only used for checking purposes. The calculation of p for the case in which the maximum speed has been reached can therefore be based on the assumption that the driver continually endeavours to keep to the maximum speed and accordingly alters the steam admission by the regulator. The regulator is completely closed only on long brake stretches and before stopping the train. The point on the line where the regulator is closed before the train runs into a station is not fixed, because when the regulator is shut depends on the judgment of the driver, on the time at his disposal to keep to the time table, the location of the signals, the visibility, etc... Inside these points the graph of the tractive

effort of the train cannot be determined with mathematical exactness.

The graph of the run of the train shortly before its stop, i. e. the braking

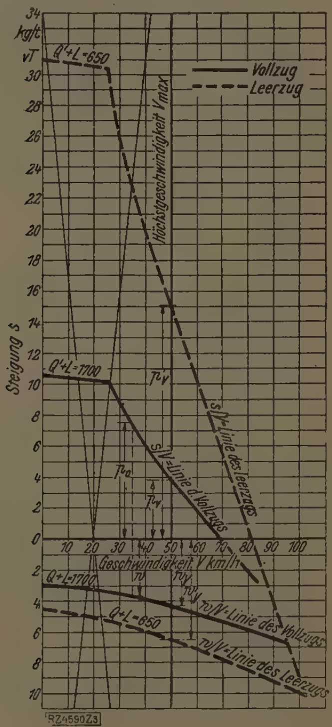


Fig. 3. — s/V and w/V diagram for 10-wheeled goods locomotive with an adhesive weight of 120 metric tons.

Explanation of German terms:

Steigung = Gradient. — Höchstgeschwindigkeit = Maximum speed. — s/V Linie des Vollzugs (Leerzugs) = s/V graph of full (empty) train.

graph *c* best plotted with the aid of a curve, forms a paradola of the 2nd or 3rd degree, according to the friction of the brake blocks being assumed to be equal for all speeds or increasing in length as the speed decreases.

A reasonable assumption is that the duration of the run with the regulator closed is about equal to that with the brakes applied.

The probable appearance of the diagram of the run and also of the tractive efforts can therefore be ascertained.

The wear due to the tractive efforts. — This wear and the additional stress on the line on up-gradients on account of the increased pull (including the starting pull) must be taken into consideration by calculating the reduction factor η_z; this is found by comparing the areas of the rolling resistances, and the tractive forces of the service line with those of an equally long, level, and straight line. Let the area below the graph of the tractive effort be called F_z, and that under the graph of an equally long, level, and straight line, F_0 (= *ABFE* for the fully loaded train, and = *A'B'F'E'* for the empty train, fig. 2). Then

$$\eta_z = \frac{F_0}{F_z}.$$

The value η_z becomes a maximum on down gradients where neither pulling nor braking is necessary, and on such sections becomes > 1.

The brake forces. — When the train has reached its maximum speed, the greatest available tractive force per ton weight of train is

$$\frac{Z}{L+Q} = p_v + w_v.$$

If the gradient $s < p_v$ or if it is negative, the actual tractive force expended per ton weight of train will vary about an average value

$$\frac{Z'}{L+Q} = (s + w_v) \text{ kgr. per ton.}$$

TABLE III.

Slip $\Delta\lambda$ and curve factor η_r for various radii of curvature.

Radius of curvature, *metres*	180	200	300	400	500	600	800	1000	1500	2000	2500
Slip $\Delta\lambda$, %	0.417	0.375	0.250	0.188	0.150	0.125	0.094	0.075	0.050	0.0375	0.030
Curve factor η_r, according to *Webb*	0.1	0.14	0.28	0.38	0.47	0.54	0.63	0.69	0.80	0.90	1
— — — according to the Pennsylvania R.R.	...	0.100	0.165	0.23	0.28	0.34	0.45	0.54	0.72	0.87	0.98
$\Delta\lambda \cdot \eta_r$, according to *Webb* . . .	0.042	0.052	0.070	0.071	0.070	0.068	0.059	0.052	0.040	0.034	0.030
— — — to the Pennsylvania R.R.	...	0.038	0.041	0.043	0.042	0.042	0.042	0.041	0.036	0.033	0.029
Average value of $\Delta\lambda \cdot \eta_r$	...	...	0.060	0.057	0.055	0.052	0.049	0.045	0.038	0.0335	0.030

If $s < - w_v = s_B$, the tractive force must be braked if the maximum speed is not to be exceeded. If the influence of the rotating masses is neglected, the braking force per ton weight of train amounts to $b_0 = - s + w$. If the length of the section on which the brakes are applied is l_B and the coefficient of friction of the braked wheels, which is assumed constant, is μ kgr. per ton, then the sliding distance amounts to

$$\Delta l_B = \frac{l_B}{\mu}(-s + w_v) \text{ km.}$$

and expressed as a percentage of the total distance:

$$\Delta\lambda = \frac{100\,\Delta l_B}{l_B} = \frac{100(-s + w_v)}{\mu}\ \% \qquad \text{(II).}$$

The average frictional force in the braking section before the train approaches its stop can be calculated as follows by means of the formula for the work done during braking, if one does not deduce it from retardation due to braking the whole train being brought to a stop by the brakes:

$$\frac{1}{2}\frac{(Q+L)\,1.06}{9.81}\left(\frac{V}{3.6}\right)^2 = \frac{V^2(Q+L)}{240}$$
$$= l_B[B + W \pm (Q+L)\,s].$$

wherein V represents the speed of the train at the beginning of the braking section in km. per hour, l_B the length of the braking section in km., W the mean train resistance in kgr. for $\frac{V}{2}$, and B the mean brake force in kgr. The formula gives the brake force per ton weight of train, viz:

$$b_0 = \frac{V^2}{240} - w_m \mp s \quad . \quad . \quad \text{(III).}$$

Putting the brake retardation equal to γ m/s^2, a similar deduction gives:

$$b_0 = \frac{1000\gamma}{g} - w_m \mp s \text{ kgr. per ton.} \quad \text{(IV).}$$

For reasons of exactness the formula (IV) for determining b_0 is to be preferred.

The wear on curves and braking sections.

The influence of the *curves* on the life is calculated with the help of the curve factor of graph *a* shown in figure 1. The effect of all curves is collectively taken into account by obtaining the mean reduction coefficient

$$\eta_{r_m} = \frac{\Sigma l_g - \Sigma l_r \eta_r}{\Sigma l_g - \Sigma l_r} \quad . \quad . \quad . \quad \text{(V)}$$

where l_g and l_r are lengths of the various straights and curves and η_r is the coresponding curve factor.

The wearing effect of the *braking forces* on the permanent way is considerably greater than that of the tractive forces of the locomotive, as is well known. An estimate of the ratio of the abrasive effects of these two forces is not possible without trials.

For this reason an approximate solution and deduction of a ratio between the wear due to the curve forces and that due to the brake forces has been used as follows.

According to figure 4 the lead or lag

$$\Delta\lambda\ \% = \frac{\Delta l}{l}\,100 = \frac{100\,s}{2R} = \frac{150}{2R}.$$

Using the graphs shown in figure 1 for the curve factor η_r the values given in table III are obtained for the various radii of curvature for $\Delta\lambda$ in %, for the curve factors η_r according to *Webb* or the Pensylvania Railroad and for the products $\Delta\lambda\eta_r$.

The mean values of the table give the following law of wear as a rough approximation : $\eta_r\ \Delta\lambda = \frac{130}{R+1880}$ % between the limits R $\gtreqless \frac{300}{2\,500}$.

It is remarkable that the value of of $\Delta\lambda\eta_r$ % increases as the radius decreases, whereas just the opposite would be expected on account of the increased sliding friction due to the guiding forces. For these reasons the formula corresponding to the observations on the Pennsylvania Railroad

$$\eta_r\Delta\lambda = \text{const.} = 0.042 \quad . \quad . \quad \text{(IV)}$$

should give more correct results.

As to how far this law is influenced by the conical form of the German tyres, which by the way is rapidly worn off, further trials alone would show. It is probable that the length of the sliding distance is of primary importance as regards the amount of wear of the rails. If this consideration is applied to the braking force, its effect can be approximately obtained if the « slip » of the wheels is ascertained just as in the case of the curves. As the revolving wheel must be brought to rest by means

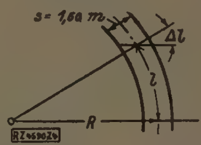


Fig. 4. — Lead and lag of the wheels in a curve.

of the brake force, the sliding distance of the wheel l_B on the braking section Δl_B :

$$\Delta l_B = \frac{l_B b_0}{\mu},$$

and expressed as a percentage

$$\Delta\lambda_B = \frac{100\,b_0}{\mu}\ \%.$$

The coefficient of reduction η_B can be calculated from the law of wear on curves according to the formula

$$\eta_B\Delta\lambda_B = C_B = 0.042 \quad . \quad . \quad \text{(VI}a\text{)}$$

on the assumption, which is not quite correct, that the wearing effect of the guiding forces on curves can be neglected as compared with that due to the wheels being stopped. The value of C_B will in reality be greater.

For the calculation of the life of the rails, the braking sections and the curve sections are summarized, by extending the formula (V) as follows :

$$\eta_{r,B} = \frac{\Sigma l'_g + \Sigma l_r\eta_r + \Sigma l_B\eta_B}{\Sigma l} \quad \text{(VII)},$$

wherein $\Sigma l'_g = \Sigma l_g - \Sigma l_B$; η_r and η_B being the reduction coefficient corres-

ponding to the curves or braking sections l_r and l_B.

Under consideration of the reduction factor for the effect of the tractive forces η_z, which has been deduced above, the average useful load of the rails of a line can be equated as follows :

$$T = \eta_{r,B}\,\eta_z\,300 \text{ million tons gross weight of train} \quad \text{(VIII)}$$

and the average life in years

$$n = \frac{T}{\text{yearly gross load in tons.}} \quad \text{(IX)}.$$

A numerical example is given in figure 2.

For the purpose of following up the wear of the rails from an economic point of view, it would be advisable not to base this on the calculated average value of the life of the entire length, but the line should be divided into sections, within the limits of which the life of the rails is either equal or very nearly so, as shown by the tractive effort and resistance diagrams. As can be seen from figure 2, the graph of the pull suddenly alters at some points. The trial rails must not be located on or near such spots as this would increase the difficulty of investigating the causes of wear.

Our knowledge regarding the scientific determination of the life of the rails is so imperfect today, that we are not even in a position to ascertain the most economical curve between two tangents by means of a satisfactory scientific method. We are more or less dependent on our own judgment and on general rules, unless the radius of the curve is fixed by special circumstances. The expansion of our knowledge regarding the relation between the economics of permanent way administration and the horizontal and vertical lay-out of the line will be followed by various improvements of existing railway lines, which are not now attempted because the economic value of such improvements can not today be proved.

[625. 2 (0]

The influence of the design and the condition of passenger carriages on their motion,

by Dipl.-Ing. Paul SPEER,
Reichsbahnrat, Berlin-Grünewald.

From *Glasers Annalen*, vol. 5, No. 1289.

The motion of railway vehicles. — General arrangement of passenger carriages. — Wheel sets and bearings. — Supporting springs. — Guiding axles. — Bogies with double bolsters. — Bogies with single bolsters. — The body. — Draw- and buffer gear.

The motion of railway carriages.

If a rigid vehicle provided with circular wheels runs on a level, straight and non-elastic track at a uniform speed, the wheel pressure does not alter as compared with what it was when at rest. The tractive effort required for the motion of the vehicle remains unchanged. All forces are in a state of equilibrium. Alternating acceleration of the masses, *i. e.* occurrences which are felt as jerks, and changes in speed of the various masses, *i. e.* shocks, do not occur. The motion will therefore be ideally « smooth » in every way.

When a railway vehicle is in motion, the conditions are, however, considerably more unfavourable. Vertical, horizontal and longitudinal forces occur which cause parasitic motions and jerks. These are mainly due to the layout of the permanent way or to the condition of the track. Now and again other influences may be the cause, such as a sudden change of acceleration or retardation of the masses as a result of a speed alteration. Characteristics of the vehicle itself cannot directly give rise to jerkiness of motion. They, however, greatly influence the effects of the disturbing forces set up. The design and the condition of the vehicle are therefore indirectly of the greatest importance as regards quiet running.

The track of the railway is not everywhere straight and even. It is elastic to a certain degree. The track must naturally adapt itself to local conditions and will therefore often deviate from the straight. As far as possible, the dynamic effects during the run are taken into consideration by suitably constructing the permanent way. In order to counteract the centrifugal force in curves, the outer rail is laid higher than the inner. This is done so that the resultant of the centrifugal force and the gravitational force is at right angles to the line joining the heads of the rails, the pressure between the wheel and the rail being in the same direction as on the straight. Centrifugal force cannot therefore cause a lateral movement of the vehicle, if its value corresponds to the available superelevation. As it is however proportional to the square of the speed, the effect of the superelevation will only be felt to full advantage at a certain speed. It is obviously not always possible to run round the curves at the exact speed corresponding to the su-

perelevation. For this reason, when designing a railway vehicle, one must take into account the fact that the resultant of the centrifugal force and the gravitational force may deviate from the vertical to the line joining the heads of the rails, and that lateral forces therefore occur when running round curves.

If the radius of a curve were to follow immediately on the straight, the centrifugal force would take effect instantaneously. A heavy lateral jerk would be the result. Transition curves are therefore provided. If these are perfect, the transition from the straight to the curve and the increase of the acceleration occur gradually up to the maximum. Similarly, the superelevation must not commence abruptly. It should be gradually led up to by means of a transition ramp. If the transition curve is satisfactory as regards construction and condition, the motion of the vehicle will be steady and smooth. Unavoidable changes of shape as a result of specially hard wear or due to the weather will, however, produce differences now and again. The nature of the ground may sometimes be so unfavourable that the layout cannot be adapted to the dynamic effects of the vehicle. It may not be possible to arrange superelevations, transition curves and ramps at all positions, *e. g.* in stations, at switch points and at crossings. The following causes will further produce forces acting on the vehicle in a lateral direction : Sudden changes in direction from the straight, running up against obstacles such as guide rails at level crossings — especially if these cross the rails diagonally —, alteration in play between the wheel and the rail on the straight, one-sided deflection of the track, etc. These forces need not, however, always be due to the contact between the wheel and the rail. They can also originate in the body of the vehicle itself. For example, the centrifugal forces when running through curves, the influences of the leading and trailing vehicles, the changes in direction of the tractive effort, all act directly on the body of the vehicle.

Certain obstacles, such as the gaps in the rails at points and crossings, the connections between the rails — the joints — are unavoidable, as they form part of the actual construction. The influence of these obstacles can be limited to a minimum by careful laying and maintenance, but it cannot be entirely avoided. Other obstacles possessing a detrimental vertical effect are not inherent to the construction, but are due to unavoidable deficiencies in the condition of the rails, such as corrugated wear, elastic deflections, etc.

Forces which disturb the motion of the vehicle in a longitudinal direction are mainly caused by alterations in the tractive force and more rarely by unequalities of the track.

The unavoidable influences occurring during the motion of the vehicle must be considered when designing it. The design of the vehicle should be such that the effect of the forces acting during the motion should be as small as possible, *i. e.* they should not be felt as disturbing shocks, or changing accelerations, or jerks. If this cannot be accomplished by the general design, separate devices must be incorporated which so convert the jerks that they are no longer felt by the passengers as being uncomfortable.

The position of the vehicle in the train can also be of importance as regards quiet running. The position at the end of the train is in itself the least favourable. However, a carriage which is soundly constructed and in good condition will also run quietly at the end. At other positions in the train special influences which do not arise in the carriage itself may cause jerky running. Carriages may, for instance, produce an unfavourable effect on other carriages in front or behind. In the front part of the train, especially in the case of the first vehicle, motions of the engine may be transferred to the carriages.

If it is required to investigate the influence of the construction and the condition of the passenger carriages, it is necessary to define « quiet » and « jerky » running. In the end, the deciding factor is the perception of the passengers which is influenced by the amount of work which their muscles must accomplish in order to balance the body against all the acting forces, so that it is in a state of rest. The body unvoluntarily assumes a position of equilibrium which requires least effort, if the conditions are such as to allow this. An unsuitable support, *e. g.* wrongly constructed seats, can force the body to assume another position than it naturally would. In order to retain equilibrium, the passenger is forced to exert muscular efforts which are unnecessary in the natural position, this leading to tiredness. This is, however, only concerned with the actual motion of the train in as far as it accentuates the effect of other features. If the body is situated in a moving railway carriage, accelerations and retardations act on its mass in various directions on account of the conditions described above. They produce additional forces or jerks, which the passenger must counteract by muscular efforts. If they assume a certain size the sensation of unstability is felt. The vehicle runs « jerkily ».

It is particularly uncomfortable if the influences are felt as « impacts ». If one body hits another an « impact » results the importance of which is dependent on the speed when colliding and on the mass. The maximum value of the force of impact is the product of the mass of the body and the effective acceleration. If the time of impact is known, the work done in impact can be calculated as follows, under the assumption that the impact forces follow a sine curve relatively to time :

$$A = \frac{m.\, b^2.\, t^2}{8\pi^2}$$

where :

m is the mass of the moving body,
b the maximum acceleration occurring, and
t the time of impact.

If the vehicle is rigid, the undiminished acceleration such as for instance that produced by impacts between wheel and rail, will be transmitted at its full value to the passengers. The work done in impact on the human body will assume correspondingly large values. As the impact acceleration can become very large and because the mass of the whole vehicle acts between the wheel and the rail, the impacts and the work done in impact would become so large that the strength of the constructive materials would be overtaxed and the influence on the passengers unbearable.

Elements of construction must therefore be employed which render the effects of the forces acting on the running vehicle harmless, before they can materialise as impacts in the interior of the vehicle. These energies must be converted into kinetic energies and destroyed in doing work, so that the occupant of the car does not feel any impacts, but only gentle movements in the form of oscillations.

The conception « jerky motion » can be divided into two main groups of phenomena :

1. actual jerks, *i. e.* the whole carriage or parts of it move in an uncomfortable manner, and

2. jerkiness in a wider sense such as disturbing noises as a result of motions (rattling, roaring, squeaking, etc.).

The motions will never be unidirectional, but will alter in direction. They will therefore always be vibrations or oscillations. Ultimately therefore all phenomena which are felt as being jerky may be summarised under the common heading of « oscillations ». Oscillations as such need not necessarily cause unstability. Quite the reverse, they are used

for converting large energies into a bearable form, by avoiding impact effects. They are, however, of a disturbing nature if the amplitudes are too large, if they repeat themselves continually at short intervals, or if large alterations of acceleration or retardation are implied, effects which are also called « jerks » (1).

It is useful in this connection to differentiate between :

1. Visible oscillations, *i. e.* real motions which are clearly perceptible by eye and can, if necessary, be measured by simple instruments. The body of the vehicle could also vibrate in this way if it were perfectly rigid.

2. Invisible, but noticeable oscillations. These are caused by the body of the vehicle not being rigid, but elastic. They are also known as « vibrations » or « roar ». They often produce noises (roaring). Ingenious devices are required to make them visible or to record them.

The oscillations may occur over the whole body of the vehicle, on some parts or only on some constructional elements. Even if the oscillations of the whole body are insignificant, some element may oscillate violently.

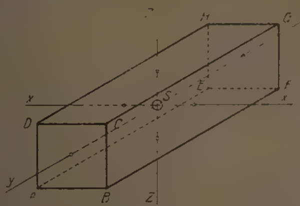


Fig. 1. — Directions of oscillations of the carriage body.

If 3 axes, X, Y, and Z are drawn through the body of the vehicle, the oscillation of the rigid body may be resolved into the following components :

1. Torsional oscillations about the X axis : pitching.

2. Torsional oscillations about the Y axis : rolling.

3. Torsional oscillations about the Z axis : rotating.

4. Oscillations in the direction of the X axis : shaking.

5. Oscillations in the direction of the Y axis : recoiling.

6. Oscillations in the direction of the Z axis : bumping.

The same components hold good for elastic or sensible oscillations.

The oscillations can occur singly or jointly. Some will, however, be more prevalent than others according to the particular conditions.

Oscillations are caused if a body capable of swinging is accelerated by means of a force. If the motion is not accompanied by losses, the swinging body will continue moving in the same manner. The amplitude and the period of the oscillations or the number of cycles will remain unchanged. Such an oscillation is called harmonic. Every body or system has a certain period of oscillation, the « natural frequency of oscillation ». In reality, however, oscillations will not take place without losses, but will always have to overcome resistances, which will gradually reduce the amplitudes, until they disappear altogether.

Of the forces which have the effect of damping oscillations those which depend on friction are of the highest importance for railway vehicles. In the presence of pure dry friction, the damping force remains of the same value during the whole process of the oscillation until the motion stops. Because, however, its magnitude is the same at all speeds from rest onwards, it can only be overcome when the occurring forces

(1) The phenomena are dealt with in detail in an essay « Der Ruck », by MELCHIOR, in the *Zeitschrift des Vereines deutscher Ingenieure*, No. 50, of 15 December 1928.

have reached the magnitude of the frictional forces. Constructional parts, which by oscillation are to convert the work done in impact into motion in order to diminish its effect, will therefore transmit forces up to a magnitude which is equal to the friction without alteration in value.

In the case of other influences which oppose oscillation the damping force is proportional to the speed of the swinging mass. The effect of such damping is far more favourable. It in non-existent in the case of rest. Because it only comes into effect when the motion begins, and its magnitude corresponds to the speed at any moment, it will adapt itself to the occurring forces and oppose them in a manner depending on their magnitude. Damping without the use of dry friction can for instance be produced by moving a piston in an air cylinder or a plate in a container filled with oil. Its use would be very desirable on railway vehicles, but it cannot be realised, as the required constructional parts could not be designed to give the required result. The main reason for this is that damping must occur on a stroke of definite length which must not be too short. This can, however, not be embodied in the parts which stand in need of it, on account of want of space or the amplitude being too small.

A further kind of damping where the opposing force is proportional to the square of the speed of the oscillating mass cannot be employed at all for railway vehicles.

If the oscillation is caused by a force which acts periodically, then it is forced. If the frequency of the driving force is equal to that of the natural period of oscillation of the swinging body, this is called critical impulse. If two oscillating systems are coupled with one another, they produce coupled oscillations.

Pure free oscillations hardly ever occur in the motion of a railway vehicle; they are in most cases compound.

When the vehicle is running on the track, oscillations are bound to occur, because the track does not consist of a homogeneous straight line, but is composed of separate joined rails with gaps between them. As each joint is passed, an elastic deflection of the track will occur. The trace of the rolling wheel will therefore show a downward deviation. Apart from this, every time a gap is crossed, a considerable impact will occur between the wheel and the rail, whose frequency will depend on the length of the rails and on the speeds of the vehicles.

The prevalent belief, that the motion becomes jerkier as the speed increases, is incorrect. It is true that, at first sight, this theory seems to be well founded. The magnitude of the impact naturally depends on the speed. The greater the speed is, the greater the effect of the impacts ought to be. The effect on the vehicle does not however depend on the impact itself, but on the work done in impact. This in turn depends on the duration of the impact and on the acceleration, according to the formula given above. As the speed inceases, the time element, however, decreases. It has not yet been fully ascertained if the deflection of the rails at the ends become smaller as the speed increases because the time intervals decrease between the separate impacts of the wheel. It seems, however, according to the latest observations, that as far as can be practically measured, the deflection is independent of the impact sequence of the wheels at the joints, and the speed. When negotiating curves the motion will become more unsteady if the centrifugal force exceeds a certain value. Furthermore, the influence of the speed on the steadiness of motion is not uniform. There are carriages the motion of which becomes steadier as the speed increases and vice-versa. There is a certain speed

for each vehicle depending on the length of the rails, which is called « critical », and at which it runs least steadily. The critical speed is that at which the forces which cause the oscillations recur in time intervals corresponding to the natural period of oscillation of parts or of the whole of the oscillating system. When designing a railway vehicle, one must therefore endeavour to avoid natural periods of oscillation which correspond to the sequence of the joints when running.

The impression of the movements of the vehicle on the passenger which can briefly be refered to as the « motion » is caused by the relation between the vehicle and the track. When designing a railway vehicle, one must reckon with certain characteristics of the latter, which may cause jerks and movements.

Constructional defects of the vehicle, which may cause unsteady motion, can be classified in the following groups :

1. Those which can be seen on the carriage ready for service;

2. Those which can only be detected after dismantling the separate parts;

3. Those which cannot be perceived with the eyes and the usual measuring devices when the vehicle is standing still.

The defects 1 and 2 may be caused by ordinary wear, by special influences in service or by faults in the constructional material. The deficiency 3 would be due to invisible strains in the body of the vehicle or to insufficient strength.

The construction must naturally be such that satisfactory conditions can be attained and maintained without difficulty. It must allow of careful manufacture within a reasonable limit of expenditure. If the occurrence of deficiencies on account of wear or special strains during service cannot be avoided, facilities must exist for them to be easily recognised and dealt with.

The construction of passenger carriages, in as far as it exerts an influence on the motion of the vehicle, will now briefly be discussed.

General arrangement of passenger carriages.

Important parts as regards the motion of the passenger carriages are the body, the draw- and buffer gear, the running parts, and the shock-absorbing elements. Of indirect importance is the brake, as it influences the condition of parts which are important for the motion. According to how the body of the carriage is supported on the running parts, one must differentiate between carriages with fixed axles, carriages with radial (flexible) axles and carriages with bogies.

The knowledge of the possibility of oscillation of the carriage body on its running parts, the natural frequency of oscillation of the body and the position of its centre of gravity are of fundamental importance for the construction of a passenger carriage, as regards its motion. In order to determine by calculation the direction and magnitude of the oscillations analytically, the moments of inertia about the three axes X, Y, and Z (fig. 1), and the radii of gyration must be ascertained. The moments of inertia of the body of a passenger carriage are more or less known. The use to which it is to be put fixes the overall dimensions — length, breadth and the height, — as well as the internal furnishings, in other words the distribution of the weight. The only latitude would be in the arrangement in a horizontal direction of the parts supported in the under-carriage such as the brake and the storage thanks. As regards the vertical height, their location too is practically fixed. The moment of inertia about the Y axis, which is specially important for the motion, cannot therefore be influenced to any extent.

The moments of inertia of the body of the car must therefore be taken as fixed and the attention concentrated on the design of the remaining constructional parts, especially the support on the running parts.

The oscillations about the Y axis and in the direction of the X axis are the least clear, and are most difficult to determine. This still leaves wide scope for the investigations of experts on oscillations. The purpose of the investigation of these motions as a basis for designs will be the determination of the natural periods of oscillation of the swinging systems, so that resonance cannot occur.

The length of the body of the coach will be given in the main by the purpose for which it is to be employed. It is economically advantageous to make it as large as possible, for as the length increases the weight per seat decreases. This is due to the fact that the weight of heavy parts, such ad draw- and buffer gear, brake, end walls and also the running parts, if the same arrangement is assumed, is independent of the length of the carriage body. It therefore becomes proportionally less as the length increases. The distance of the supports of the carriage body on its running parts, *i. e.* the distance between the axles on carriages with guided axles, and between the pivots on cars with bogies.

The part of the carriage body which protrudes beyond the bogie pivot support is called the « overhang ». When running round curves the end of the centre line of the vehicle departs from the centre line of the track because of the overhang, by a distance dependent on the ratio of the whole length to that between the supports. This side movement can be further increased if the carriage body can slide sideways on account of play in various parts, and it can differ in direction with two coupled vehicles. It must not exceed a certain value in order that the areas of the buffers, which are limited in size overlap sufficiently all the time. For this reason, the overhang must not exceed certain maximum values for a given distance between the body supports. They have been ascertained and specified for the usual lengths of carriages for various dimensions of the buffer areas. The maximum values of the overhangs are therefore fixed with regard to the length of the carriages. It is worth considering however if it is not advisable to choose smaller values because the overhanging masses can exert an effect on the motion by reason of their dynamic influence. In this connection the moment of inertia about the Z axis is of great importance. It does not depend so much on the actual length of the overhang as on the mass incorporated in it. If this is not excessive and if heavy parts can be arranged in the vicinity of the Z axis the influence of the overhang on the motion will not be too unfavourable.

The dynamic effects of the mass incorporated in the overhang will be specially noticeable when the body is deflected from its direction of motion. This occurs at changes of direction on account of the tractive force transmitted by the draw bar and through the guiding of the wheels along the rails. The dynamic effects will therefore be most pronounced when running into and round curves. The actual guiding of the vehicle should be effected by the wheels. It would be best if the deviation caused by the wheels and by the tractive force would begin simultaneously. The latter is however due to the leading vehicle and therefore occurs before the wheels run into the curve. The desired condition may be more closely approached the nearer the leading wheels are placed to the end of the vehicle, *i. e.* the longer the wheel base is compared with the length of the body of the carriage.

In the case of bogie carriages, the width of the body depends on the distance between the bogie centre pins.

This distance will be limited to a maximum value, as one cannot go below a certain minimum width of stock. At a given distance between the pivots, the total length of the wheel base can be increased if this is desirable on account of the length of the overhangs, by extending the wheel base of the bogies. Apart from this the most suitable distance apart of the bogie axles can be obtained from the following considerations :

There is play between the rails and the flanges of the wheels, which becomes larger in curves according to the radii, through what is known as « gauge widening ». The sets of wheels are therefore able to execute movements in the direction of their axes besides rolling along. The plane of the flanges can also assume an oblique position as regards the direction of the rails. The angle which is thus formed is known as the striking angle. The play, as such, of the wheel pairs in the track is advantageous, as it enables them to roll freely and without flange friction. If the section of the tyres still retains its double conical shape, which it had when new or after leaving the repair shop, the set of wheels will readjust itself to the central position when it has been displaced to either side by a jerk. If a set of wheels departs from its central position on account of a jerk, it will overshoot this position when again returning on account of the centering forces. The set will accomplish lateral oscillations which will gradually fade away unless new influences occur. The oscillations obey pure sine laws as Boedecker deduced geometrically for the first time in his treatise « *Die Wirkung zwischen Rad und Schiene* (*The relations between wheel and rail* ») (1887). If, however, the wheels are cylindrical, no readjusting forces will act on the set of wheels. The motions will solely be directed by the influence of the rails. If such a wheel runs up against a rail with its flange, it will rebound again on account of the elastic impact. The conical form of the tyres alters its shape after being in service a short while because of wear taking place. Temporarily a more or less cylindrical shape will occur. This is followed by the formation of a shallow groove. It has been ascertained by observations that the wheel sets of bogies actually perform a sinusoidal side to side motion. This is, however, largely influenced by the construction and condition of the permanent way and specially depends on the elasticity of the rails when acted on by lateral forces. The elastic readjustment forces due to the track appear in the main to cause the sinusoidal motion, whereas the actual form of the rolling surface of the wheels is of secondary importance.

The sinusoidal oscillations, generally speaking, have but little influence on the riding qualities of the carriages. Experience indicates that the form of a well worn tyre of a bogie possesses a certain resemblance to the rail head. This points to the fact that the wheel only departs from its central position by small amounts, or in other words, that the movements in the direction of its axis are small. The amplitudes of the sinusoidal oscillations are consequently very small. The flanges generally show very little signs of wear. This leads to the conclusion that they seldom touch the rails. As has been ascertained by observations, this generally only occurs when negotiating curves of a radius of less than about 800 m. (40 chains), especially when running into them and apart from this only when larger obstacles are met. Experiments have also shown that, other conditions being equal, the motion of a carriage remains unaltered if it is equipped with proper conical, cylindrical or well worn wheels. It therefore makes no difference to the motion of a passenger car whether the tyre is new or worn. The calculation of the sinusoidal motion of a set of wheels on the

assumption of the proper conical rolling surface does not therefore furnish a useful basis for the design.

The dimensioning of the wheel base is, on the other hand, of importance as regards the motion. The greater the wheel base is, the less is the tendency to skew. On the straight the striking angle is the smaller the larger the distance between the axles. It does not in this case affect the motion to any considerable extent however, as it is very small in itself. The distance between the axles will therefore be of very little account when the vehicle is running on the straight. The sinusoidal motion is not influenced by the distance between the axles as is apparent from the above. The conditions are however different when negotiating curves or running into them. The distance between rigid axles which results in the smallest angle of skew in the range of R = 400 m. (20 chains) to R = 800 m. (40 chains) lies between 3 and 4 m. (9 ft. 10 1/8 in. and 13 ft. 1 1/2 in.). If the radius is less, the most favourable value is below 3 m. The range from about 3.10 m. to 3.20 m. (10 ft. 2 in. to 10 ft. 6 in.) is best avoided, as it roughly corresponds to one revolution of a wheel of a diameter of 1 m. (3 ft. 3 3/8 in.). Any defects in the wheel, even if they are small, may set up resonance on account of simultaneous impulses with the joints and thus have a detrimental effect on the motion.

Rigid axles may be used if their distance apart is less than 4.50 (14 ft. 9 in.). If it is greater, pivoted axles must be employed. The simplest and most satisfactory construction of adjustable sets of wheels, the guiding or flexible axle, will be described later.

Wheel sets and axle boxes.

The arrangement of the wheels which is most usual on road vehicles, *viz.* where the wheels revolve round fixed pivots fitted to the body of the car, is not practicable on vehicles running on rails. The wheels must transmit such large horizontal forces to the frame that it cannot be guided by a fixed pivot with sufficient safety. Special cases excepted, the wheels of railway vehicles are joined by an axle so as to form a set of wheels. The load can be suitably supported on the axle in such a way that lateral jolts can be transmitted satisfactorily by the bearings.

The wheel sets roll along the rails and are therefore the first constructional parts to be subjected to jolts which they transmit. Their construction and their condition are therefore of particular importance as regards the motion. The wheel sets form the most important part of the unsprung mass. Their weight is of capital importance with respect to the work done in impact. If good spring gear is provided, the total axle load is of comparatively little importance in relation to the weight of the unsprung mass. The wheel sets must therefore be designed as light as possible. The widespread opinion that a high axle load, as such, is a deciding factor on the smoothness of the run is incorrect; in fact it is in contradiction with the observation that particularly heavy carriages run more smoothly than light ones, which may of course also be due to the more favourable design of the springs which is possible in the case of the heavy vehicles.

As the separate parts of the whole set are not perfectly rigid but naturally possess a certain, if small, ability to take up work elastically, they also contribute to rendering harmless the work done in impact. The axle shaft is elastic to a certain extent and the ends of the axle are deflected. This must be considered when designing the bearings. The tyres and the ends of the axle must be perfectly round and concentric in order to prevent any disturbing movements when running on the rails. The tread circles of the wheels forming the set must be

of the same diameter. Any unequalities in the distribution of the weight of the revolving masses should not exceed certain maximum values. Otherwise jerky oscillations in the direction of the Z axis will occur. Tyres which are not perfectly round *i. e.* such possessing flats also frequently cause disagreeable vibrations of the body of the carriage, which are felt most just over the faulty wheel set and to a greater or lesser extent all over the vehicle. Non-circular wheels can further be recognised by knocking or rattling noises. The wear of the tyres must not exceed the maximum permissible value. Flanges run sharp as such do not cause jerky running, but are positive proof that something is not in order with the wheel set, which may exert a detrimental influence on the running. It particularly indicates that the wheel sets run askew.

It is important for the motion that the normal position of the axles is exactly parallel to one another and at right angles to the longitudinal axis of the frame to which they are fitted. The centre of the distance between the tread circles of the set of wheels must lie on the longitudinal axis of the carriage. It is important, for this reason, that those parts which exert a directional influence on the set of wheels be made and fitted with great accuracy. If guiding stresses do not occur, the guides of the axle boxes determine the position of the wheel sets. They must in this case be very exactly machined. Any possibility of deflection on account of the load must be considered when machining. If any set of wheels is askew in its mid-position the flanges will become badly worn, which can lead to sharp edges being formed. The resistance to motion will be increased. Forces will occur which cause unsteady motion.

It has not yet been clearly established what degree of play between the bearing and the axle collar is detrimental to the motion. According to the experience at present available, bearing play does not exert an unfavourable influence on the motion. Even if the journal bearings have been fitted exactly, a certain amount of play soon develops on account of wear, which, however, shows only a small increase subsequently. The author is not aware of any instance where the riding qualities of carriages could be improved by taking up any bearing play that may have been there before. On the contrary, observations were made which seem to show that a certain amount of bearing play helps to alleviate lateral jerks.

A fairly large clearance between the axle box and the axle box guides can only have detrimental effects in such designs as make no provision — such as for instance supporting the load on springs which exert re-centering forces — for satisfactorily guiding the axle. At all events, large play due to wear should be looked for when judging the quality of motion of a carriage. If it occurs one can definitely conclude that certain deficiencies of the running gear are causing unsteady motion. The chief causes of excessive wear are springs which are too weak, incorrect position of suspension, wheel sets which are askew, etc. A bad load distribution of the body can also act in this sense.

Supporting springs.

The jolts which are caused by the wheel rolling on the rail must be rendered harmless by converting them into work causing an elastic deformation. All constructional parts of the vehicle and of the permanent way possess a certain amount of elasticity. The capability of doing elastic work which the wheel sets possess has already been mentioned. The roadbed too is not perfectly rigid. Part of the work done in impact is therefore taken up by unintended ef-

fects. This part is however comparatively small and its magnitude cannot be exactly determined. A special constructional element must therefore be provided, whose main object is to absorb the impact work. This is the purpose of the supporting springs. They very often however have to perform additional duties, such as exerting a directional force on the wheel sets in order to keep them in their mid-positions or to re-set them again if they have been deflected therefrom. In some arrangements they also transmit forces due to lateral movements of the body. As the springs are the only elastic link which can alter its vertical dimension to any appreciable extent, they determine the vertical height of the support and accordingly the horizontal position of the body. The construction and the condition of a part which is of such importance for the quality of the motion must, of course, be absolutely perfect.

The kinds of supporting spring which are employed on railway vehicles are laminated or leaf springs and helical springs. Leaf springs are particularly prevalent. They possess the advantage that they can be more easily and suitably accommodated than helical springs. and that they are more adaptable for different purposes.

The aim should be to treat leaf springs as a beam of uniform strength, fixed at the centre, with the load suspended from both ends, in order to achieve the greatest capacity for taking up work combined with least expenditure of material and uniform stress over the entire length. For calculation purposes only one half of the beam, or one arm, is considered. The usual formulæ are therefore based on a cantilever of uniform strength whose plan is an isosceles triangle. The base of the triangle is formed by the clamp or support. The load is suspended from the apex.

Because the beam would become too large for mounting, it is cut into separate strips, called « leaves », which are piled on one another. This results in the leaved triangular spring, whose line of deflection is a circular arc.

For a given force P the dimensions of the spring depend on :

the deflection f
and the strain σ_b

The deflection f is indicative of the quality of the spring, *i. e.* the softness of the motion, as will be discussed later. It is limited by constructional conditions. The total maximum permissible deflection of the spring is limited by the permissible range between the highest and the lowest position of the buffers, due to the weight of the empty body, the load and an addition for impact deflection.

The permissible strain σ_b is limited by the characteristics of the material. It must not exceed a maximum value, even if the distance between the stops is fully used, *i. e.* when the whole range of deflections comes into play and the spring rests on the stop.

For a beam of uniform strength the following formula holds good :

$$f = \frac{6\ P.\ l^3}{n.\ b.\ h^3\ E}$$

$$\sigma_b = \frac{6.\ P.\ l}{n.\ b.\ h^2}$$

where l is the length of one arm of the spring, E is the modulus of elasticity, n is the number of leaves, b is the width and h is the thickness of the leaf section.

For a spring of given dimensions :

$$f = C.\ P.$$

The constant C is an index of the softness of the spring. The larger it is, the greater is the deflection due to the same load and therefore the softer is the spring.

The reciprocal $c = \frac{1}{C}$ is called the hardness of the springs or the spring constant.

If this is substituted in the above formula, we obtain :

$$f = \frac{1}{c} P.$$

The constructional design of leaf springs for railway vehicles, however, involves modifications of the pure triangular form and of the assumptions contained in the formulæ. If these modifications are not taken into account, the calculated deflection and strains will not correspond to those actually obtained. Similarly the line of deflection will not be a circular arc. The modifications will be briefly discussed in the following.

The lever l of the bending moment, which in the formula was taken to be the fixed length of the spring arm, varies its mangnitude as the deflection increases. The triangular form of the plan cannot be exactly adhered to. The edges must be cut away where the spring is clamped at the support in order to enable the leaves to be held together, and at the end in order to enable the load to be applied. The leaves should be piled one on the other so that they are entirely relaxed when unloaded and touch over their entire surface. This cannot, however, be complied with in reality. The leaves are even purposely submitted to an initial stress so that the ends do not lift and separate when the load is applied. With the present methods of manufacture, it is furthermore hardly possible to obtain a full contact of the surfaces. The forces are therefore transmitted mainly by the ends of the leaves. They must therefore not terminate in points, as the specific pressure would be too high in them. Generally several leaves are continued right up to the end of the spring, *i. e.* they are made as long as the leaf above. The area of the section will not be quite a rectangle, as was assumed in the formulæ. The narrow sides and the edges are rounded off. Ribs and grooves are provided in order to prevent the leaves from moving sideways. Further deviations are due to wear, to overheating and to tolerances in manufacture. Because the leaves do not all lie in one plane but are piled one on the other, the radii of curvature become less from leaf to leaf. If the spring is bent beyond the straight position, the conditions are reversed, *i. e.* the lower a leaf is placed the higher it is stressed.

The leaf springs are generally loaded in such a manner that in addition to the vertical forces, further forces occur, which act in a longitudinal or in a lateral direction. Additional forces are produced by pressing the leaves together in the buckle. Further uncertainty in actual practice, as compared with the calculation, is due to the friction which occurs between the single leaves on account of their relative motion when deflected.

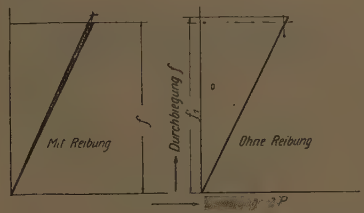


Fig. 2. — Leaf spring with and without friction.

Explanation of German terms :

Belastung = Load. — Durchbiegung = Deflection. — Mit Reibung = With friction. — Ohne Reibung = Without friction.

In order to study the influence of leaf friction let us consider a spring possessing friction. It has a load-deflection curve as shown in figure 2*a*. The hatched area represents the part $2A_r$ of the

work done, which has been converted into frictional work during the compression and the release of the spring. During compression alone the following work is done :

$$\text{by change of shape } A = 2.\,P\,\frac{f}{2} = P.\,f$$

$$\text{by friction } \frac{2\,A_r}{2} = A_r.$$

In the case of a spring without friction that part of the work A_r which was absorbed by the real spring in friction during compression, will also be converted into change of shape. Under the same load 2P, the spring will therefore be further deflected to f_1, as shown in the load-deflection graph, figure 2*b*. The whole work will be absorbed in change of shape :

$$A_1 = P.\,f_1.$$

The following relation therefore holds:

$$A_1 = A + A_r$$

A_r can be calculated according to Marié (*Hütte* I, p. 661). The ratio of the part of the work absorbed in friction to that converted into change of shape is :

$$\frac{A_r}{A} = \zeta$$

$$\zeta = \frac{\mu\,(n-l)\,(n'' + 0.5\,n').\,h}{n.\,l}$$

μ is the coefficient of friction;
n' is the number of non-bevelled upper leaves.

$$A_1 = A + \zeta.\,A$$
$$= A\,(1+\zeta)$$
$$P.\,f_1 = P.\,f.\,(1+\zeta)$$

When springs are calculated according to the triangle formula, friction is neglected, as mentioned above. The value f_1 is therefore obtained. The deflection of the real spring, influenced by friction, is :

$$f = \frac{f_1}{1+\zeta}$$

The value μ is dependent on various factors, mainly, however, on the state of the surface of the leaves and on the degree of lubrication.

Furthermore, in those parts which hold the spring, friction arises tending to oppose deflection. Its value, too, cannot be accurately determined, and can therefore not be considered with certainty in the calculation. The design should therefore be so chosen that this friction cannot occur to any appreciable extent.

The main causes of the differences of the actual deflections from calculated results are the alteration of the lever arm during deflection, the longitudinal force caused by the load suspension, and friction.

The values obtained from the triangle formula are only suitable for an approximate calculation of the stress. If greater accuracy is required, the actual arrangement of the superimposed leaves and the real cross section must at least be considered. The stresses caused by lateral and longitudinal forces should be checked for safety. Their value, as compared with the stress due to the vertical load, will however not be very high generally. One need not therefore be afraid of fully utilising the springs for purposes where they are subjected to additional pure tensile or lateral bending stresses. It is rather more risky to subject the leaves to longitudinal compressive stresses.

The effect of the helical spring depends on a steel rod being twisted by the influence of a force. Only rods of square or round section are used on the passenger cars of the Reichsbahn (German State Railways). Other Administra-

tions also employ elliptical, rectangular or other sections. In order to accommodate the rod, it must be wound into a helix. Nowadays cylinder-shaped springs are mainly employed, whereas formerly they were also made in cone or even barrel form. The elasticity which compares with the deflection in the case of leaf springs and the stress can be calculated, for rods of circular section, by the following formulæ :

$$\text{Deflection } f = \frac{64.\ P.\ n.\ r^3}{d^4.\ G}$$

$$\text{Stress } \sigma_d = \frac{16.\ P.\ r}{\pi.\ d^3},$$

where P is the active force, r the radius of the coil, d the diameter of the rod and G the sliding modulus.

It follows from

$$f = C.P$$

that helical springs behave in the same manner as leaf springs, as regards elasticity. The formulæ are similar for other rod sections. The equation $f = C.P$ also holds for these. The values C and c have the same significance for helical springs as for leaf springs. If on account of limited space more than one helical spring must be used, or if the diameter of the rod would become too large in proportion to the diameter of the coil, several helical springs are often arranged within one another. The single concentric springs are of equal height when unloaded. In order that all the springs wear equally, they should be so dimensioned that they are equally stressed. In order to achieve this, the rod diameters d should be so dimensioned that they are proportional to the radii of the coils r, *i. e.* the following condition must hold :

$$\frac{r_1}{d_1} = \frac{r_2}{d_2} = \frac{r_3}{d_3} = \text{constant.}$$

It is a disadvantage from the point of view of springing the vehicle that the load-deflection curve follows a straight line. As has been mentioned, the smoothness of the motion depends on the magnitude of the value f. As this, however, increases as the load becomes greater, a fully loaded carriage should run more smoothly than an empty one. The greater the useful load is compared with the dead weight, the harder an empty carriage will run. As regards springing, it is therefore advantageous if a carriage has a high dead weight and few seats. For a heavy carriage, it is therefore easy to design the springs in such a way that it also runs softly when empty or only partially full. This cannot however be achieved so satisfactorily with the ordinary arrangement of springs in the case of vehicles possessing a large capacity and small dead weight. It was therefore proposed to build springs with equal resiliency at all loads. This would be possible for leaf springs for instance, if the effective length of the spring is reduced during loading by he use of suitable stops. This would, however, require great exactitude in the shape of the spring, which cannot be realised with railway carriage springs. For goods wagons a compromise has been effected by arranging two leaf springs one above the other. If the wagon is empty, only one is in action. When this has been fully deflected by the load the second spring comes into effect. This arrangement can however only be used for large load differences. The transition from the working of one spring to that of the next is too sudden. As the passenger load only varies in very small steps, this arrangement cannot be used for passenger vehicles.

If the deflection of helical springs were to be graduated in order to adapt their smoothness to the load, several concentric springs, of varying height when unloaded, and coming into action

in succession, would have to be used. The load-deflection curve would be a broken line instead of a straight one. For reasons of constructional order, this arrangement has so far not been resorted to on those of the railway vehicles on which helical springs can be used. In fact, the dead weight, which has to be borne by the springs as well, represents such a large proportion of the total weight in the case of passenger carriages that the available part of the spring travel would be insufficient for conveniently subdividing it in order to obtain a gradual deflection.

The effect of a spring is due to the fact that it bends or twists under the action of an impact. It therefore absorbs the work of impact. When this has been accomplished it expands again, returning part of the absorbed work to the mass forming the load, which again returns it. The spring therefore absorbs it anew, returns it again and so on. On account of the elasticity of the spring the work done in impact is therefore employed in producing oscillations of the mass forming the load. The force of impact is therefore no longer felt as such in the interior of the vehicle.

The working capacity of a spring, *i. e.* the work which it absorbs under the influence of a force, is equal to the integral of the force over the distance it travels, *i. e.* the resilience. According to figure 2, therefore :

$$A = \frac{P.f}{2}.$$

If in the calculation of a triangular spring P is the force acting on one arm of the spring, this value only gives half of the work capacity of the spring. For the whole spring :

$$A = P.\, f.$$

The more a spring is deflected by a given force, the greater will be the work absorbed, and the « softer » is the springing therefore.

A combination of the formulæ for f and σ of leaf springs results in the following important relation :

$$f = \frac{l^2.\, \sigma_b}{h.\, E}$$

The stress at the maximum load should be chosen according to the maximum permissible value for the particular material. Assuming this to be a fixed value, the formula can also be written :

$$f = \text{const. } l^2.$$

For a given load the resilience increases with the square of the length of the spring. The longer the latter is, therefore, the softer it will be. This fact also appears from another relation. Indeed, the working capacity of a spring can also be expressed by the following equation :

$$A = \frac{V.\, \sigma_b^2}{6.\, E},$$

wherein V is the volume of the spring.

For a given stress this can also be expressed as :

$$A = \text{const. } V \text{ (1)}.$$

The greater the volume of the leaf spring, the greater is its capacity of doing work. The volume, however, increases with the length of the spring. The spring, therefore, possesses the most favourable dimensions if it is just deflected to the maximum extent permissible, in the available space, by the maximum load.

In the absence of opposing influences, a spring loaded with a mass would oscillate continuously with the same amplitude, after having been accelerated by an impact, the period of oscillation depending on the magnitude of the loading mass and on the deflection, *i. e.*, on the

(1) This value also holds similarly for helical springs.

dimensions of the spring. The natural period of oscillation is :

$$t = 2\pi \sqrt{\frac{f}{g}}$$

wherein f is the deflection under the load P and g is the acceleration due to gravity. The natural period of oscillation is also given by :

$$t = 2\pi \sqrt{\frac{m}{c}}$$

wherein m is the mass loading the spring and c is the spring constant. The period of oscillation therefore depends on the deflection or the hardness

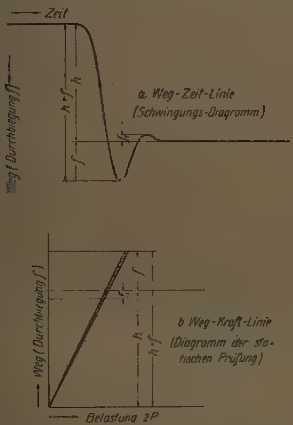


Fig. 3. — Leaf spring diagrams.

Explanation of German terms:

Weg (Durchbiegung) = Deflection. — *a*) Weg-Zeit-Linie... = Time-deflection curve (oscillation diagam). — *b*) Weg-Kraft-Linie... = Load-deflection curve (static test diagram).

of the spring, on the magnitude of the mass loading the spring and on the dimensions of the springs.

In actual fact, however, the amplitude of the oscillations becomes gradually smaller until the mass again comes to rest. Figure 3 shows the oscillation (time-deflection graph) of a leaf spring 2 m. (6 ft. 6 3/4 in.) long, as used for railway vehicles. It was loaded with a mass equal to the part of the carriage weight carried by the spring. A slab attached to the ends of the spring was used as the load. It was first raised and then allowed to fall. Damping caused especially by friction between the leaves is very great as shown by the oscillation diagram. In figure 3 the conditions of the oscillation are transferred to the diagram of the static test (force-deflection graph). The energy of the fall is absorbed by the spring. The spring is deflected from its original form at the beginning of the test by a distance $h + f$. Now h is due to the pure static load of the weight of the slab, so that only that part of the energy which corresponds to the deflection f is available for conversion into energy of oscillation. This however is returned by the spring when it rebounds and is converted into work expended in raising the weight. The slab (or the body of the carriage) is lifted by the spring when it returns, by $f + f_1$. Compared with the « Position of rest under load » the potential energy G f is now available. When falling down into this « Position of rest under load » the energy is returned to the spring, which in turn converts it into work done in changing its shape. The initial quantity of energy is thereby absorbed.

For railway vehicles it is very desirable that spring supported parts should come to rest quickly. For continuous oscillation, especially with large amplitudes, can have very disagreeable effects, even if hard jolts or jerks are not felt and the carriage runs quite « softly ». Springs which are too soft or which possess too little damping can easily give rise to long continuous oscillations in the direction of the Z axis, and can cause rotating movements round the Y axis or

pitching about the X axis, if the jerks do not occur simultaneously and with equal force on all springs. If jerks act only on one wheel of a pair, a rotation round the Y axis will be set up. The body of the carriage then swings as though it were a kind of « hinge » which could also be described as being a « spring hinge ». Forces which act directly on the body can also give rise to oscillations of this « spring hinge » type. In this connection the height of the centre of gravity is of importance, as will be shown later.

If the springing is too hard, due to too small a deflection, *i. e.* too small a working capacity or especially on account of excessive friction, it gives rise to short disagreeable jerks in the direction of the Z axis which are felt as vibrations. This is also referred to as hard or rough running. The whole body of the carriage oscillates in a short and disagreeable manner. If the available space for the springs is less than the amount of the deflection when oscillating, the springs will hit their stops, which results in disagreeable jerks. If this only occurs rarely and when the carriage is unusually heavily loaded, it can be accepted. If, however, it is of frequent occurrence even at the ordinary loads, the car must be considered objectionable. In this case either the dimensions of the spring are too small or their condition is faulty.

With leaf springs damping is greatly assisted by friction between the leaves, caused by relative movements of the single leaves on deflection. Disagreeable continuous oscillations of large amplitude are therefore avoided. There is, however, the danger of the friction becoming excessive, resulting in hard running. Then the springs will only come into action when the change of the load is greater than the opposing frictional force between the leaves. This deficiency can, however, be successfully reduced by greasing the springs between the leaves or also by employing leaves of large section, thus minimising their number and the consequent friction surfaces. It would naturally be preferable to grease the springs regularly while in service compared with only doing so during shop inspections. This would however hardly be possible.

With helical springs, damping is comparatively small. There is hardly any friction as there are no moving parts sliding on one another. The spring will therefore already act under small changes of load. If they are used as main springs the car will run very softly. The amplitudes of oscillation, however, easily become large and continuous and can then be very uncomfortably felt. One could be led to think that the oscillations could be damped by arranging concentrically several helical springs possessing different natural periods of oscillation. However, as can be seen from the relations given above, this is not possible, if the springs possess equal resilience. One would therefore have to use several concentric springs, which would not be of equal heights in the unloaded state, so that the value of f under load and at the same time the period of oscillation should differ. As has however been mentioned in another connection there are constructional reasons against such an arrangement

Both kinds of spring therefore have their respective advantages and drawbacks. One therefore endeavours to employ the two kinds in series, if this is possible on account of the constructional features. The leaf spring is allowed the greater part of the admissible springing and the helical spring the smaller part. The lighter jerks which the leaf springs transmit without oscillating thanks to their friction, are then eased by the helical springs. As regards the running, the leaf springs need not be greased or lubricated in this arrangement. This is however advantageous even in this case, as other benefits result. The life of the

spring is increased and especially the uncertain calculation factor introduced by dry friction is dispensed with as regards fitting the spring, where the knowledge of the deflection, as it determines the height of the body, is of particular importance.

It is obvious from these considerations that next to the construction, the condition of the spring is of the utmost importance for the running of the carriage. All similar springs of a carriage must possess the same characteristics, *i. e.* their force-deflection graphs and their oscillating properties must be equal. If this is not the case, disagreeable torsional oscillations of the body will be set up about all three axes. The period of oscillation of the system must naturally be very different from the impulses due to the joints in the rails.

The dimensions of the springs can in most cases easily be tested. The space available for the deflection must in all cases be equal. If this is not the case the body of the car will not be level. Either the condition of some of the springs is not perfect or the distribution of the load is uneven. If faults in the dimensions of the springs are detected, these must by all means be rectified. One must of course make sure beforehand that the carriage is standing on a level track. If the latter is inclined, the springs will be deflected more at the lower end of the carriage than at the higher end. Important characteristics from the point of view of oscillation, such as for instance the amount of friction, cannot be seen or measured when the built-in spring is at rest.

Radial axles.

The simplest railway carriage is the carriage equipped with two « rigid axles ». Leaf springs, from whose ends the body is suspended by means of spring eye-bolts, are supported on axle-boxes which are held in axle box guides without any play. This arrangement is only suitable if the distance between the axles is small, as has been discussed before. Above a certain value, curves can no longer be easily negotiated. Lateral jerks will be transmitted to the body at full strength, if one neglects the inconsiderable work done in bending some of the constructional parts and the oscillation on the springs round a point lying between them, *i. e.* round the « spring hinge. ». Passenger cars with fixed axles are therefore rarely to be found in service.

It is almost universal practice now to allow the wheel sets to depart from their central position, if required by the run of the rails. Fixed ranges of play which are determined by calculation and experience are arranged between the axle boxes and the axle box guides in a longitudinal and lateral direction. The axle box guides now only serve the purpose of limiting the deflections of the wheel sets to the maximum permissible values, when specially severe jerks induce a large deviation. At first it was thought necessary to provide some means of ensuring a relative adjustment of the wheel sets. In the case of carriages with 3 axles the end axles too were adjusted by the centre axle. Such arrangements have the disadvantage that all axles are adjusted in that moment when the adjusting axle begins to run into the curve, when therefore the end axle is still running in the straight.

These « coupled » radial axles were therefore universally replaced by the « free » radial axle, which is considerably simpler and cheaper in first cost as well as in upkeep. In the latter case, a leaf spring is used for exerting a directional influence on the wheel set by means of a suitable suspension of the body. It allows the wheel set to be deflected from its centre position by external influences, but again gently returns it to its original position when the cause of the deviation disappears. The spring must offer a certain opposition to

any deflection so that the set does not obey the smaller forces which continually occur. It would in this case always oscillate from side to side causing unsteady running. The readjusting or centering force is caused by the inclination of the spring eye-bolts by which the body is connected to the springs. The action is shown in figure 4. The spring eye-bolts are equally inclined both ends if the wheel set is central. The ends of the spring are therefore subjected to the horizontal forces H, which are equal in magnitude on both sides, but opposite in direction. The spring is consequently under tension, but a directional force tending to influence the central position of the wheel set is not set up. If on the other hand it is deflected from its position by a distance a, by some external influence such as for instance in the direction of the arrow, the inclination of the spring eye-bolts is also altered. It is increased on the side R, and decreased on the side L. The horizontal forces acting on the springs will alter accord-

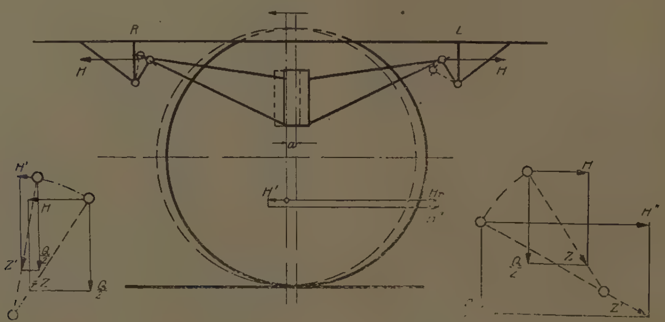


Fig. 4. — Radial axle.

ingly. On the side R, the force decreases, becoming equal to H′, and on the side L it increases to H″. A directional force $H_r = H'' - H'$ now acts on the spring opposite to the direction of the arrow, and becomes gradually smaller until the wheel set has again returned to its central position. The inclination and the length of the spring eye-bolts are of the greatest importance for the smooth running. The centering force must not be too great, because the wheel set otherwise would be held too rigidly in its normal position. It must not be larger than the friction between wheel and rail. The effect as a radial axle would otherwise be jeopardised. If on the other hand the centering effort is too small, the wheel set will also be deflected by comparatively small forces. In consequence the set would continually oscillate from side to side.

The simplest carriage equipped with radial axles possesses 4 wheels. The load is supported on the two axles. The axle load is therefore equal to half the total weight of the carriage if the load is evenly distributed. It is therefore larger than that on carriages of the same construction but possessing more than two axles.

As is shown in figure 5, each axle transmits the whole amount of any differences in the track level to the body. The effect of the jerks must be wholly absorbed by the springs of this one axle. The carriage with only two axles will therefore not satisfy really severe requirements regarding the quality of the vertical motion. It will nevertheless run

satisfactorily at speeds up to about 80 km. (50 miles) per hour, if in good condition.

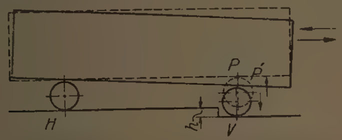


Fig. 5. — Motion of carriage with two axles.

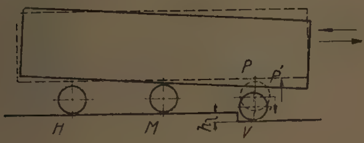


Fig. 6. — Motion of carriage with three axles.

As regards the absorption of vertical jerks, the carriage with three axles has the advantage. As is shown in figures 6

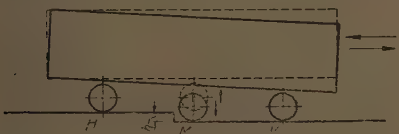


Fig. 7. — Motion of carriage with three axles.

and 7, the body of the car is always additionally supported by the springs of a further axle whenever one of the wheels has to overcome differences in track

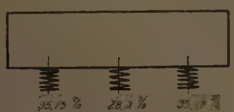


Fig. 8. — Load distribution of three-axle carriage.

level. In order to ensure a definite guidance by the end axle, the middle axle as shown in figure 8, should be loaded to a smaller extent than the end axle. Its springs are therefore made weaker. This aim will however only be attained if the deflection of all the springs exactly corresponds to the predetermined loads. If the springs of the centre axle are set up too high under no load conditions, or if they possess deflection characteristics different from those of the end axles, they may under certain circumstances take a larger share of the load than the latter. The only actual effect of the smaller dimensions will be that the stress is greater in this case. The result will be unsteady running. This deficiency cannot be recognised by sight, as it is not possible to judge how high the springs are when the load is removed and what deflection characteristics they possess. In order to form an opinion, one is entirely dependent on the effect when running.

Carriages with three flexible axles are mostly suspended on simple leaf springs. In spite of their advantage with respect to the absorption of vertical shocks as compared with the two-axled carriages, they do not satisfy the very highest demands as to quality of running. The vertical action might be considerably improved if it were possible to lubricate the leaf springs regularly during service. It is, however, very doubtful if such lubrication could be accomplished economically and with the required thoroughness.

If the suspension of the load of the carriage with radial axles is suitable, the effect of the lateral forces will be eased to a certain extent by the oscillations of the carriage body. Lateral forces will therefore not wholly be noticeable as shocks, but will produce oscillations of the body, which will be bearable if they are of the right character. The length and inclination of the spring eyebolts are very important also from this point of view. Apart from this, lateral jerks are only eased by the body oscillating on what we may call the hinge « center » formed by the springs.

The pivot for the motions round the « spring hinge » is formed by a point, which lies between the two lateral groups of springs, or by a corresponding axis. The spring constant being known, the moments of the acting forces can be calculated. From these and the moment of inertia the period of oscillation of the spring hinge can be ascertained.

In the case of carriages with radial axles it is most important that all spring eye-bolts are of the same length. Their inclinations must be equal in the mid position of the wheel sets, and they must be of the right amount. The springs must all be the same size.

It should be noted in this connection that the inclination of the eye-bolts also depends on the deflection of the springs. The greater this is, the greater the inclination of the eye-bolts will be.

As one had to content oneself with simple leaf springs, a limit was set to the possible improvement in the running of carriages with radial axles. Useful suggestions for some kind of double spring were made, and were occasionally tried. The constuction however, involved the assembly of a large number of parts.

The most important advantages of the radial axles, their extreme simplicity, would therefore be considerably reduced. If value is attached to particularly smooth running, both in a vertical and horizontal direction, bogies must be used.

Bogies.

General arrangement.

As regards the effect of jerks originated by the rolling of the wheel on the rail, bogies as such already offer an advantage on account of the way in which the load is supported. As is shown in figure 9, when an obstacle is run over amounting to a difference of height h, only half of it, *i. e.* $\frac{h}{2}$ affects the body. A greater advantage however is that there is considerably more freedom in arranging the springs than with radial axles. More exacting demands can therefore be complied with. Groups of springs can be arranged in series, each of which possesses special properties for easing

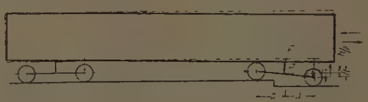


Fig. 9. — Motion of carriage with bogies.

shocks and oscillations, without the use of complicated constructional parts.

Bogies may be regarded as small independent carriages, on which the body of the car rests. They can therefore move underneath it in a horizontal plane, *i. e.* they can revolve.

As compared with the whole length of the carriage, their wheel base may be relatively small. Fixed axles can therefore be employed. A directional force by means of springs, supported on the axle boxes need not necessarily be provided.

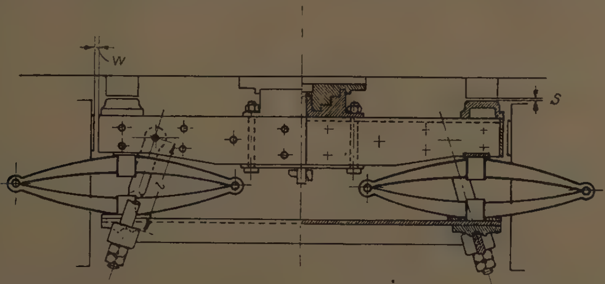
In order however to save the axle box guides and the axle boxes, it is to be recommended that this possibility be not entirely put aside.

On account of the comparatively small wheel base, bogies can accommodate themselves well to the conditions of the track, especially when running into and through curves, and also when the carriage does not run true on the straight. On the other hand, the total wheel base of the carriage may be very great.

A lateral shock on a wheel will set up a rotary movement of the bogie. The energy of the shock will already be partly converted into kinetic energy, thus rendering the shock more or less harmless. In order to enable the bogie to turn freely, the total load of the carriage body is supported by a central pivot — apart from special designs for exceptional cases —. In order to ensure this requirement being fulfilled, a clearance of at least 0.5 mm (0.0197 inch) is provid-

ed between the side friction blocks, as shown in figure 10. If the side friction blocks are loaded, friction will occur, which will only allow rotary motion when the effective forces are strong enough to overcome it. As long as they are smaller, they are transmitted without diminution to the carriage body. Although they cannot attain a large value, they will produce disagreeable effects such as vibrations or harsh riding.

A suitable suspension of the carriage body on the bogie frame is a good way of easing the action of lateral forces. Lateral forces which originate in the body itself, *e. g.* centrifugal forces, mass effects when the body is inclined, forces transmitted by the leading and trailing cars, can be well absorbed by this method. In this respect, it is easier to vary the design than would be the case with a carriage with radial axles. Hence one is at liberty to arrange for oscillations — of the kind as were found to be satisfactory for running — to be set up by lateral forces, and thereby prevent them from being felt as shocks. Lateral forces originating from the wheels cannot be directly transmitted from the bogie frame to be body of the carriage, provided the body is suspended by means of swing links. These forces however



S = Play between friction blocks. —. W = Bolster side movement. — l = Length of swing link.

Fig. 10. — Two-piece bolster.

will serve the purpose of moving the body out of its central position, thereby slightly raising it. The necessary work will be supplied by the lateral force. Therefore this force will not appear as a shock.

As is shown in figure 10, the body rests on the bolster which in turn is hinged into the bogie frame by means of links. Knife edges support the bolster by means of saddle pieces. The amplitude of the oscillation is termed the play of the bolster, and it is limited by suitable stops. When the carriage is standing on a level and straight section of track, the distance between bolster and stop should be equal on both sides. If the two distances are unequal, this must be put right, since it shows that there is some imperfection unfavourably influencing the riding qualities, such as different length of links, or faulty distribution of the load.

The body reaches its maximum amplitude of oscillation if the lateral force becomes equal to the re-centering force. While swinging to its extreme position, energy has been accumulated in the body, which is now employed in reversing the motion. Unless there are opposing forces, the body will swing back across the central position, and reach an amplitude of equal value on the other side. This would continue, *i. e.*, the body would swing backwards and forwards,

with always the same amplitude, unless there were forces to oppose it. It is therefore essential to provide some means of returning, quickly and without shock, the body to the central position, each time it has been displaced. It is important not to allow the amplitude to attain too large a value in the first instance. Therefore the opposing forces, also referred to as the re-centering forces, should not be made too small.

On the other hand, they should not — in particular for small amplitudes — be made excessive, as this would tend to produce oscillations of a jerky nature. These forces should be small at first and should grow rapidly as the amplitude increases. This increase must be steady and must not undergo sudden alterations which would lead to jerks. It is easily possible to comply with the said conditions by suitably designing the swing ling suspension and giving it a suitable length.

In the past spring checks were frequently used for this purpose, which produced the resulting force, *e. g.* by compressing a helical type spring. This method is not suitable. A bolster spring check could only serve its purpose if it would be inactive as long as the forces remained small and only slight oscillations took place.

Only if the forces increase should they be met by an increasing resistance. In other words, the re-centering force should not become effective before the bolster has made a certain movement. And then the force should not be constant; the force-deflection curve should be flat at the beginning and rise more and more sharply towards the end. It is obvious that the narrow space available and the short travel of the bolster will make it impossible to realize all requirements with spring checks. On some carriages in service, it was found to be necessary to insert the spring with a considerable initial tension. Thus even in the central position a considerable resistance was exerted, which immediately and very rapidly rose in proportion to the movement, Hence the carriage body was not permitted to begin its motion until the lateral accelerating forces had reached such an extent as to exceed the initial spring tension.

This amounted to lighter shocks being transmitted without any damping. These light shocks, however, will also occur with the carriage running perfectly smoothly, and they would without the presence of the bolster check springs be felt in the interior as nothing but a hardly noticeable lateral oscillation. With check springs an unpleasant vibration would be felt. When the accelerating forces exceed the initial tension, the spring would be compressed and accumulate work. Little of this work being absorbed, it will be suddenly released whenever the accelerating shock ceases and the spring loses its tension. Consequently the bolster will be jerked back. It will pass through the central position and compress the spring on the opposite side. This cycle will be repeated.

This shows that sudden oscillations may be the result of fitting spring bolster checks.

Another type of bolster check employing friction as an agent for absorbing lateral forces is equally unsuitable. It only becomes effective after a shock has reached such a large value as to overcome the friction. Smaller shocks would be transmitted undamped. The best device, so far as is known, would be one the action of which is based on damping without friction. For instance pistons moving in an air- or oil cylinder could be considered.

There are however considerable difficulties to be overcome, the most important being the very limited available travel.

In fact, for a correctly designed bolster suspension, a well constructed bogie and the whole carriage in proper condition, such a device is not required. An

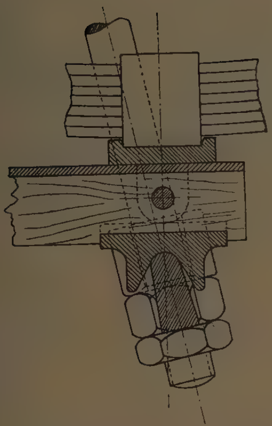


Fig. 11. — Inclined link with blunt knife edge.

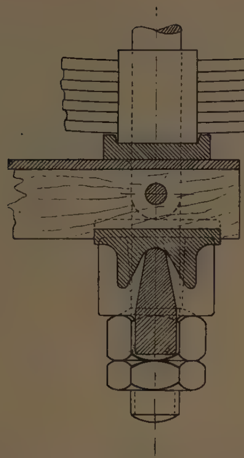


Fig. 12. — Vertical link with sharp knife edge.

ample « bolster travel », *i. e.* the amount by which the bolster is allowed to swing to either side until reaching a stop, should be provided for every carriage. Hence with the carriage in proper condition, the stops will only be reached very exceptionally, and even then the impact will be so gentle that it will hardly be felt in the interior of the vehicle.

The length of the swing links has great bearing upon the value and the characteristic of the re-centering forces. It is therefore obvious that all the swing links of one bolster and if possible throughout the whole carriage should be of equal length. Otherwise the forces counteracting an oscillation would be of different values, and unpleasant lateral movements of a jerky nature would result.

A certain resistance to oscillations is also exerted by the friction of links and knife edges. Frequently « sharp » knife edges are employed. Their bearing radius is considerably less than that of the saddle piece resting on the top. This type only offers little friction. Numerous trials have shown that « blunt » knife edges seem to be more favourable for running. In this case, the radii of the edge and the saddle are nearly equal and sufficient, though not excessive, friction occurs during oscillation. Figures 11 and 12 illustrate sharp and blunt knife edges.

If a blunt knife edge is correctly shaped, a suitable counterforce will be obtained, without the friction having any disturbing effect. This fact, it would seem, is explained by the absence of forces smaller than the knife edge friction.

Comparing links of equal lengths, inclined ones will produce a larger counterforce than vertical ones. This is due to an inclined link already possessing a certain re-centering force in the central position on account of its inclination as such, and to this force exceeding that of a vertical link, which only exerts this force when swinging out. In actual fact, the re-centering force due to the

oblique position is nearly the same for inclined as for vertical links. With the inclined type the re-centering forces on both sides, although large in themselves, act against each other and only their difference will be effective, this difference being nil in the central position. On the other hand, vertical links will have smaller individual resetting forces on both sides, but acting in the same direction they will assist one another. The real

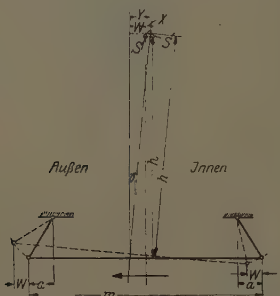


Fig. 13. — Action of bolster with inclined link.

Note. — Aussen = Outside. — Innen = Inside.

reason for the larger re-centering force of oblique links is shown in figure 13. When a bolster suspender by inclined links is displaced from the centre, it will assume an oblique position. Hence if the carriage body swings to the outside when negotiating a curve in the track, it will become inclined towards the inside. Hence when an oscillation occurs in the direction of the arrow, the centre of gravity of the carriage body will be displaced from its original position S in a direction opposite to the mention, *i. e.* towards S′. Hereby an influence is brought to bear on the body counteracting the force causing the oscillation, with a re-centering effect. The springs on the outside would be stressed more on account of the superelevation. The oblique position of the body, however, and also the displacement of the centre of gravity to the inside, act in an opposite direction, thus relieving the outer springs.

There is a further consequence due to the oblique links.

Shocks acting on the link suspension upset the balance of the forces by producing unequal forces in the links. As shown in figure 14, an additional compression will occur in the one and an additional tension in the other link. These additional forces produce a motion about the spring « hinge ». Any oscillation of the carriage body will be

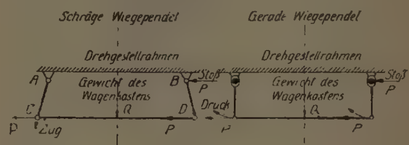


Fig. 14. — Force diagram of shocks acting on the swing link spring gear.

Explanation of German terms:

Drehgestellrahmen = Bogie frame. — Druck = Pressure. — Gewicht des Wagenkastens = Weight of carriage body. — Gerade Wiegependel = Vertical links. — Schräge Wiegependel = Inclined links. — Stoss = Shock. — Zug = Pull.

accompanied by torsional oscillations at the spring « hinge ». Therefore the kinetic problems involved are not quite easy to grasp. For the first time they were thoroughly treated by Dr. Hoenig in his investigations on the conditions governing smooth running of bogie carriages for fast trains (1910).

The scope of the present article only permits a short reference to these phenomena to be made. The total movement of the body is composed of the oscillation of the bolster, and of a rotation about the spring « hinge ». For each of these individual movements there is a momentary centre of motion. If the bolster is in its mid-position, this centre is formed by the intersection of the extensions of the two links, *i. e.* in the centre line of the body. When the bolster is out of its mid-position, the point of intersec-

tion of the directions of the links is displaced, and so is the momentary centre of motion. It appears that the result of this is an oscillation which cannot be expressed by simple formulæ. The pivot point of the spring « hinge » will be between the two groups of springs, and will not alter its position with respect to them. For the combined motion there is a resultant centre of rotation at every moment. None of the two individual

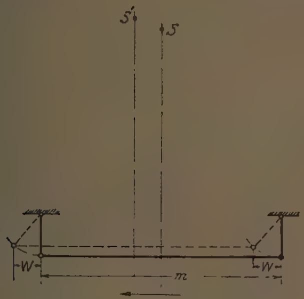


Fig. 15. — Action of bolster with vertical links.

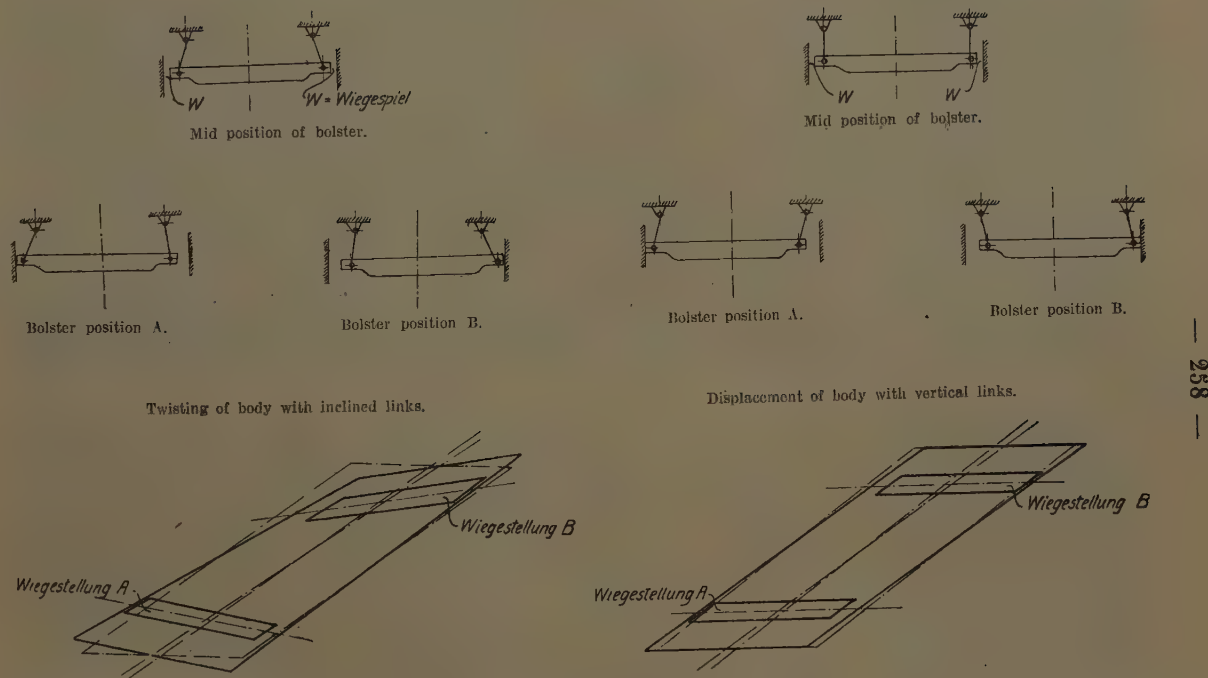
rotations can occur independently since the whole arrangement is such that the second movement is bound to occur as a result of the first. In order to calculate the two movements, one has to put into equilibrium all forces and moments occurring during the motion, including those caused by the acceleration. As a result of such a calculation, it can be proved that there are two oscillations of different frequencies, possessing separate poles, which superimpose.

One of the poles will be found to lie above the centre of gravity, the other near the centre of rotation of the spring « hinge ».

The conditions are much simpler if the bolster making the oscillation is suspended by vertical links. The load on the springs will then not vary when the bolster leaves its mid-position. Hence no influence due to the spring « hinge » will be observed. Only the centrifugal force will cause an uneven distribution of the load on the springs, when the carriage negotiates a curve. The body will incline towards the outside. The outer springs will therefore receive an additional load which is very undesirable as such, as it lessens the inclination of the body towards the inside of the curve, which is aimed at by the superelevation in order to make the resultant of gravity and centrifugal force perpendicular to the line joining the two rail heads. This deficiency can however be reduced to a reasonable value, if the extent of the bolster bearing on the springs is made as large as possible, as this distance determines the re-centering force of the springs.

With vertical links a lateral movement will not bring the bolster into an oblique position. As shown in figure 15, the centre of gravity moves by the same amount as the bolster swings out, *i. e.* from S to S′ in the direction of the force causing the oscillation.

Thus no force opposing the motion will be the result of the position of the bolster. Hence, assuming links of equal length, and the same kind of suspension, the reaction against an oscillation will be less for vertical links than for inclined ones. In consideration of this fact inclined links used to be employed in the past. Their advantage however is outweighed by a greater disadvantage which consists in the oblique position they impart to the body when swinging out. As shown in figure 16, the carriage body will consequently be distorted if the two bolsters happen to move in opposite directions, and unpleasant shocks and vibrations may result. If bolsters suspended by vertical links swing out of their mid-positions in opposite directions, the plane of the frame will remain as before and the carriage body will not be subjected to torsion. The above disadvantage would not therefore apply to this design.



Explanation : ———— Position of body at mid-position of bolster.
—— . —— Position of body at displacement of bolster.

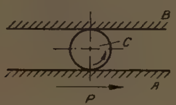
W = Wiegespiel = Bolster play. — Wiegestellung = Bolster position.

Fig. 16. — Comparison of the action in inclined and vertical links.

Endeavours have been made to obtain the same reactive forces against oscillations, employing vertical links, as occur with inclined ones. A simple way of achieving this would be to make the links shorter. Inclined links should not be too short in order to prevent the oblique position of the carriage body and the consequent disadvantage from becoming excessive. Vertical links only raise the body by an amount depending on their length. The shorter the links are, the quicker does the reactive force resulting from their inclined position increase.

Hence the play of the oscillation will be smaller, the shorter the links are made. The length of vertical links may be made considerably shorter than that of inclined ones, since the carriage body is not distorted when vertical links are used. Consequently it will be possible to obtain the same effect of total reaction against an oscillating movement with short vertical links as with oblique inclined links of greater length.

A further advantage is offered by ver-



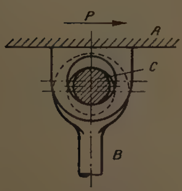


Fig. 17. — Othegraven hanging of swing links.

tical links. They allow a certain design of suspension to be adopted, known as the « Othegraven » system, from the name of its inventor. This system is based on the following considerations : A mass B, as shown in figure 17, is supported by a roller C, resting on a support A. If the latter is given a push P, this will not be transmitted to B, but will cause A to roll along under B, by means of the roller C. This fact can be usefully applied for the purpose of softening shocks. If A be the bogie frame and B the carriage body, then the effect of shocks will be reduced by this mechanism. Naturally provision must be made for the mass B, *i. e.* the body to be returned to its mid-position without a jerk, when the influence of the shock has ceased. For this purpose, Othegraven has therefore made the shape of the support A and the bearing surface of the mass B circular, as shown in figure 17.

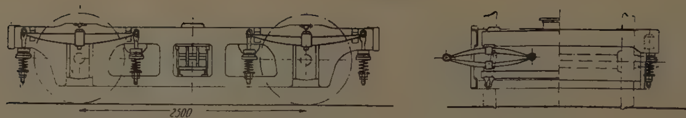
The bolt C, moving along an arc of circle when a shock occurs, simultaneously lifts the carriage body suspended on the links B and returns it into its mid-position when the force acting on it has ceased. As shown in figure 14, no unequal forces or their consequences can result in the links from a shock acting on their spring gear.

Proper working is conditional upon all bolts of the same bolster having the same direction, which must be exactly at right angles to the direction of oscillation. In order to make them self-aligning, they are supported in ball bearings.

Bogies with two-part bolster.

The bogie shown in figure 18 — the German standard bogie — is widely used throughout Europe. Two short leaf springs rest on the axle boxes, and small helical auxiliary springs are inserted between the leaf springs and the frame. As is shown in figure 10, the bolster consists of two parts, the top and bottom spring planks.

Both are elastically connected by transverse leaf springs, three or four of which are placed together to form a set. Thus there are three groups of springs



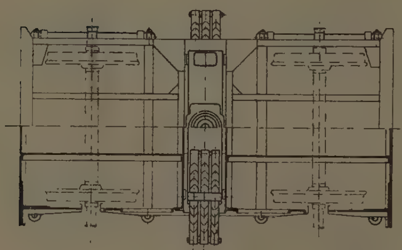
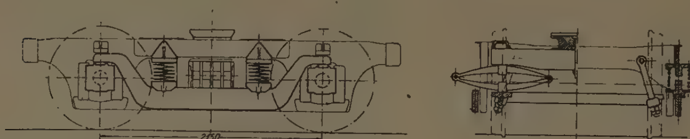


Fig. 18. — German standard bogie.

in series. The top bar carries the center casting and side friction blocks. The bolster swing links are suspended from transverse frame members. The saddle pieces supporting the knife edges and the bolster are mounted on the lower spring plank.

Generally speaking, the riding quality is satisfactory, provided the bogies and the carriage body are in perfect condition.

As the result of American experience it was however believed that a different arrangement would lead to considerably smoother running. From these considerations, the bogie known as the American type illustrated in figure 19 was developed. The frame and the bolster are not materially different from the standard design. The arrangement of



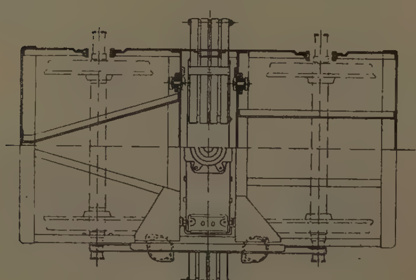


Fig. 19. — American bogie.

the springs between the frame and the axles was novel. In order to attain smooth running, the whole of the space available for the side bearing springs was utilised for fitting helical springs, and leaf springs with their great and undesirable friction were avoided altogether. There was no space above the axle boxes for the accommodation of helical springs of suitable dimensions. Therefore they had to be arranged at the side of the axle boxes. In order to support the springs, it was necessary to arrange a bar between the two axle boxes which was called the « Swan neck bar » (arch bar) from its characteristic shape. The purpose in view was actually achieved, and carriages so equipped run very smoothly.

Carriages with both standard and American bogies run smoothly provided

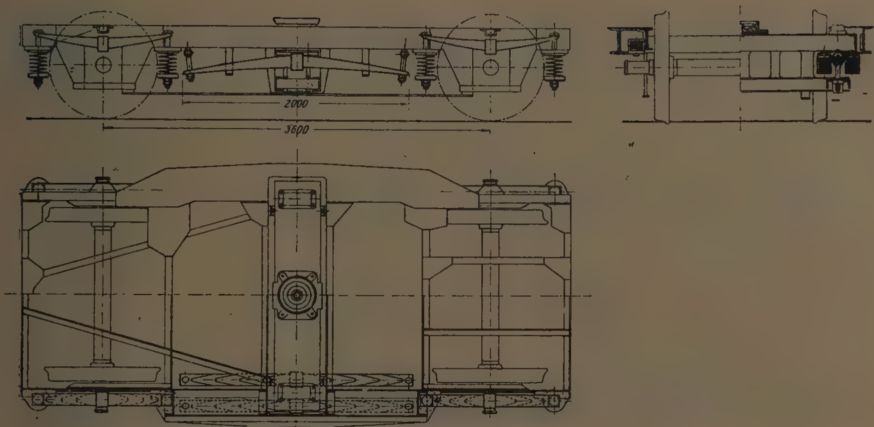


Fig. 20. — Bogie type « Görlitz II heavy ».

all their component parts are properly dimensioned. There is a certain difficulty in the upkeep, as a result of the arrangement of the bolster springs. They are not adjustable, and therefore very limited tolerances must be specified for their height. If the height of the springs in a group is not uniform, the highest springs will absorb a higher force, and the shortest a lower force than the third spring. The height of the carriage body and its horizontal position depend on the dimensions of the spring buckles, and on their distance apart. Hence these dimensions too must have narrow tolerances. Although it has been possible to comply with these requirements by means of careful workmanship, an inducement existed to investigate the possibility of a design which tended to facilitate assembly by the adoption of adjustable springs, of course without prejudice to the running qualities. The result of these investigations was the construction of bogies with one-piece bolsters.

Bogies with one-piece bolsters.

The object aimed at necessitated the use of simple leaf springs, arranged in the longitudinal direction of the bogie frame. In view of the load to be handled, and in order to obtain a simple suspension, two springs were provided on each side member of the frame. As is shown in figure 20 a cross piece rests on their buckles, the former being fitted with a bolt. From this bolt the links are suspended. The springs are suspended from the bogie frame by threaded bolts, which are prevented from moving laterally.

The bolts being independent of each other, each spring can be separately adjusted to the proper height. Therefore greater tolerances are permissible as regards their length than with springs of a two-piece bolster. As the load is suspended midway between a pair of springs, each will bear half the load. Hence the stresses will also be equal, even if the spreads differ. A number of express train carriages have already been equipped with the bogie illustrated in figure 20, which is known as the « Görlitz II » type. As their running was satisfactory, a further development of this type of bogie seemed to be advisable.

With a view to further facilitate the

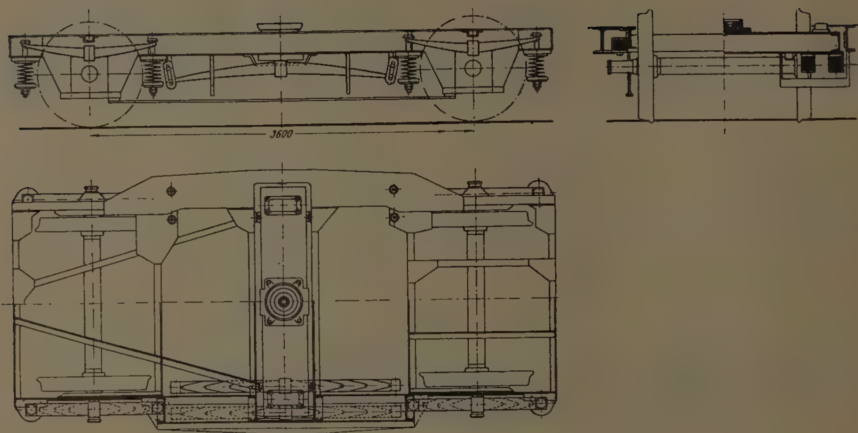


Fig. 21. — Bogie type « Görlitz III heavy ».

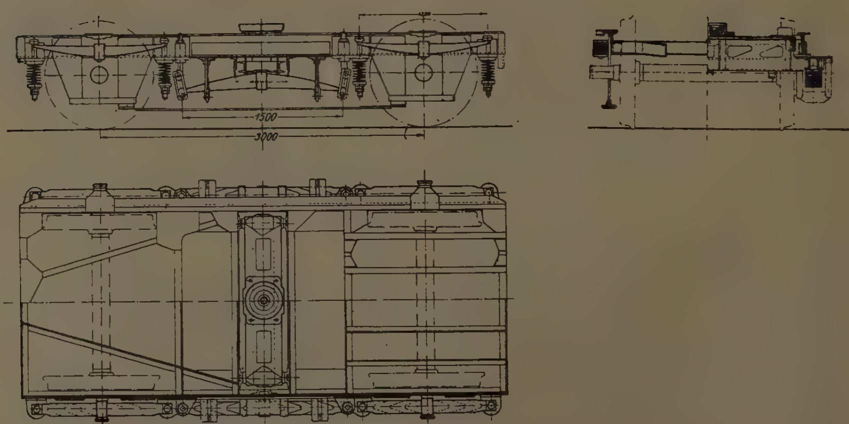


Fig. 22. — Bogie type « Görlitz III light ».

upkeep, the type shown in figure 21, and known as the « Görlitz III heavy » type was designed.

The bolster springs are suspended in shackles, which permit lateral oscillation and act as links. The shackles are suspended from blocks, whose height is adjustable. Thus with this arrangement too it is possible to adjust the bolster springs individually. A simple girder, carrying a center pivot bearing and side blocks, is laid across the two pairs of bolster springs. The carriage body rests on this girder in the usual way.

In this design the links and their accessories which were rather inaccessible,

were eliminated. According to the experience so far gained, it would appear that the design complies with the requirements as to smooth running.

As a modification of this design, a similar type « Görlitz III light » was developed, for use on four-axle carriages for ordinary passenger trains (see figure 22). The lower stresses permitted the design to be further simplified. Instead of a pair of springs one spring is sufficient for each side. The smaller height of the longitudinal frame girder and of the lower bolster girder made it possible to arrange the latter below the former. The bolster spring could be attached in such a favourable position that torsional stresses on the frame are kept very low. No transverse girders are required for taking up the vertical load.

The carriage body.

The best bogie equipment in perfect condition is not sufficient to achieve smooth riding. The design and condition of the car body must also be suitable.

The position of the center of gravity is of particular importance. Its vertical axis is determined by the load distributed and can readily be found by means of weighing equipments. Hence it is possible to detect and to eliminate, by proper measures, any deficiencies of the car body due to unsuitable location of the vertical axis of the centre of gravity. As shown in figure 23, the tractive effort Z acts along the longitudinal axis of the carriage, whereas the resistance W which it has to overcome, acts about the centre of gravity, in the opposite direction.

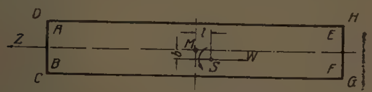


Fig. 23. — Influence of position of centre of gravity in the horizontal plane.

Therefore, the distance b between the longitudinal carriage axis and the centre of gravity must not be excessive, as undesirable torsional forces would otherwise result from the action of the tractive effort. Also with regard to the distribution of the load, the distance b should not be made too large, as the distribution of the load on both sides, and accordingly the compression of the individual springs, would be influenced. If the load is unevenly distributed in the case of two-piece bolsters, it is therefore necessary to provide springs with a height varying with the load on them. The carriage body would otherwise assume an inclined position and there would be the danger of this being adjusted by altering the lengths of the links, thus making the latter unequal. The distance b should be as short as possible also from the point of view of oscillations. If it is excessive, the frequency of the groups of springs on the two sides will differ considerably, since it will vary with the load, the spring constant c remaining the same. Accordingly, undesirable oscillations around the Y-axis are likely to occur, if the load is not equal on both sides, *i. e.* if the distance b between the longitudinal axis of the coach and the centre of gravity is large. These oscillations would even occur if the shocks were equal on both sides. As far as this is possible during construction, a correct position of the centre of gravity should be aimed at by suitable design and distribution of the load. If balancing cannot be accomplished in any other way, compensation by means of counterweights should be resorted to. This, however, is not desirable as it would mean so much dead weight.

The height of the centre of gravity is of equally great importance. mainly with reference to the origin and the characteristics of the lateral oscillations set up in the spring « hinge ». The influence of the height of the centre of gravity in the event of a vertical shock on one of the wheels is shown in figure 24.

Initially a moment will act around a centre marked O in figure 24. S is the centre of gravity of the complete carriage.

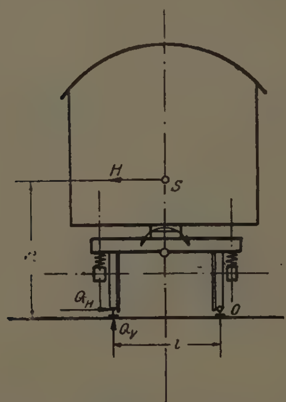


Fig. 24. — Influence of height of centre of gravity.

The forces can be expressed by the following relations :

$$1.\ Q_v \cdot l - H \cdot h = 0$$
$$2.\ H - Q_H = 0.$$

The same relations also apply if the origin of the forces is a centrifugal force H or a lateral shock Q_H.

If a vertical force Q_v of a certain value acts on the rail, the resulting force H will be the smaller, the higher h is, *i. e.* the higher the centre of gravity is situated. If, contrarywise, the lateral force H originates in the carriage body, being caused by a centrifugal force (this also acts in the centre of gravity) while the carriage is negotiating a curve, the momentum H.h and the force Q_v will be the larger, the larger h or the height of the centre of gravity is. For a lateral force Q_H acting on the rail, the momentum H.h and the force Q_v will also be the larger, the higher the centre of gravity.

Therefore, a low centre of gravity will be advantageous as far as horizontal forces are concerned, and a high centre of gravity will be favourable as regards the influence of vertical forces.

The same considerations apply to the forces on the springs and the torsional oscillations about the Y axis, dependent on these.

For a high centre of gravity, horizontal forces will have a strong effect, vertical forces only a light one, and vice versa.

Accordingly a low centre of gravity will only be favourable as far as lateral forces are concerned, *i. e.* in particular for negotiating curves. For all vertical forces caused by the track, a high centre of gravity is preferable. These facts were confirmed by numerous tests. It is an etablished fact that carriages possessing a frame embodying specially heavy parts, *i. e.* with a low centre of gravity, do not run so well on a straight line as otherwise similar carriages without these heavy parts. Similarly, carriages with a high centre of graviy tend to run less smoothly on curves than on the straight.

Additional stresses in the carriage body, other than those due to the vehicle's own weight or load, *i. e.* stresses set up during the construction or resulting from hard service conditions may have a very detrimental effect on the running. Such stresses can be explained by some parts of the carriage being assembled with an inherent stress, *i. e.* parts which do not quite fit together, being forced into position. It is also possible that certain parts have suffered a permanent deformation during service operations, whereas others have remained within their elastic limits. Carriage bodies affected with such stresses may under certain conditions of loading or if otherwise subjected to forces, be bent or twisted in a non-uniform manner. Up to a certain stress they are comparativ-

ely stiff and unelastic, but show a sudden deformation when the forces increase further. When this deformation has occurred, they are once more comparatively stiff. The distribution of the load on the supports will naturally change. When running, such carriages will show a tendency to perform sudden deflections or torsions or heavy vibrations in the direction of the X axis. These stresses are all the more objectionable as they are invisible and cannot be detected with the existing testing and measuring devices when the carriage is standing still. They also offer no possibility of a mathematical investigation, and are therefore to be avoided altogether. The construction and the maintenance of railway carriages must, therefore, be carried out carefully and with great precision.

Hence carriage bodies, which are in perfect order, must not be affected by internal stresses. Their stiffness must be uniform and sufficient to limit elastic deformations to a minimum.

Such natural vibrations of the carriage body in consequence of deflections are initiated by shocks. If these shocks occur at intervals corresponding to the natural frequency of the oscillations of the body, the vibrations can become very powerful. They are particularly noticeable in the middle of the carriage and become weaker towards the ends. These phenomena are particularly objectionable, as they cannot be mathematically determined. They do not even possess the same magnitude on all cars of the same design. This proves that they too are liable to be largely influenced by the condition of the body which depends on the assembly or on external influences. One can only depend on the investigations made on the completed vehicle.

It is characteristic of these phenomena that their maximum strength is felt in the middle of the vehicle, as has been mentioned above. Frequently they are observed on carriages of otherwise excellent riding qualities. They can also produce the impression that the carriage is subjected to longitudinal jerks. When again building cars of a type on which these phenomena were previously observed, it is possible to prevent them by altering and especially strengthening some of the component parts. It may possibly also be effective to reinforce the pillars of the body sides. Instead of increasing their cross section, it may be sometimes advisable to increase their number. One must further investigate whether the rivetting of the cover sheets and the fixing of the upper and lower braces is sufficiently good. Alterations of existing vehicles will in the majority of cases only be possible at great cost. They may be dispensed with if the phenomena do not occur continuously, but only rarely under certain conditions, and if the running of the vehicle is still sufficiently smooth, so as to make the vibrations bearable.

If the vibrations are originated in the running gear they will either be of equal strength over the whole length of the carriage body, or will be specially noticeable at isolated spots above the part generating the vibrations. If for instance the load is resting on the two friction blocks instead of on the centre bearing, or if the spring arrangement of a bogie is unsuitable, this will be felt most strongly above the defective parts. It is then easy to remedy the trouble.

When new carriages are built, a certain body stiffness must be assumed which can be secured in practice, as experience has shown. In older cars, however, and particularly in those possessing wooden body framing, deformations may occur which can detrimentally influence the running qualities. In most cases this will be due to the loosening of some component parts of the frame. The phenomenon occurs if there is a well maintained and strong frame, but if the joints of the body and particularly

those of its sides no longer meet the requirements. There will be a continuous movement of the centre of gravity, and accordingly of the load distribution. As a consequence oscillations in the direction of the X-axis and rotary movements round the Y-axis will occur. The only remedy would be a thorough repair of the body framing. Similar effects are due to a deformation called

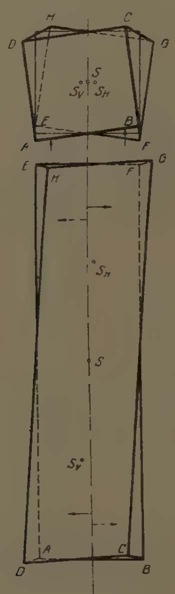


Fig. 25. — Skewing of the carriage body.

« skewing », shown in figure 25. Even if the joints of the frame are not quite good in order, the effect may be the same as referred to above.

It has also been observed on old type steel carriages, in spite of all the joints being sufficiently strong. Here the reason was found to be the insufficient transverse bracing of the car body. The deficiency was easily removed by suitably strengthening it. Whether this deficiency of the carriage body is actually the cause of the uneven running may be ascertained by checking the deformation. Frequently wooden parts are liable to produce unpleasant squeaking and groaning noises.

Certain phenomena may convey the impression of uneven running, even if the carriage is actually running well. These are mostly noises caused by the vibrations of component parts, such as rattling of doors, squeaking of windows, or their counterbalance weight, intermediate partitions, gangways between the cars, draw bar plates, play in links, loose brake shoes, etc. The cause is always in these cases a defective condition of the parts, such as loose joints, bad fixing, too much play in sliding fits, etc.

The remedy is very simple.

Draw- and buffer gear.

Very unsteady running can be caused if the draw- and buffer gear is in bad condition.

There are two principal arrangements for transmitting the tractive force. In the one the drawbar is elastically fixed to the headstock. The frame of a particular vehicle must therefore transmit the whole of the tractive force required for its own motion and that of the vehicles following it. This arrangement as such is advantageous, as the tractive force pulls the vehicles of the train, thus opposing lateral movements. Also as regards the strength of the frame no difficulty would be experienced. The dimensioning of the drawbar springs is however difficult. It must be so dimensioned that it can take up the whole tractive force of the locomotive. This however diminishes from vehicle to vehicle. For the last carriages, the spring on account of the very small forces acting on it, for its size, would be insufficient. This fact may have very unpleasant results when starting or changing speed. For this reason another arrangement is chosen.

A continuous drawbar is provided, to which each carriage is separately linked through a spring. The draw spring need therefore only transmit the tractive force required for the carriage to which it is fastened. It is therefore equally strained on all carriages and can consequently be suitably dimensioned.

Among other uses, the buffers should bear against each other in order to damp lateral movements. The drawbar equipment must therefore be ajustable in length. It must be tightened so far that the buffer springs are subjected to an initial compression, which positively ensures their contact. If one of the buffers becomes jammed when in the compressed position it can give rise to very unsteady running. If this is the case, the carriage will only bear against the other on one side, as shown in figure 26.

On account of the play a available on the other side, the carriage will tend to perform a rotating movement in the direction of the arrow. The movements

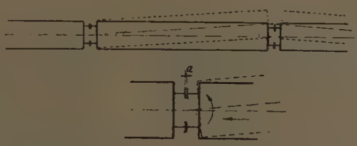


Fig. 26. — Effect of jammed buffer.

can become so strong as to involve following vehicles.

The drawbars and buffers especially the latter, can indirectly lead to unsteady running, if parts sliding on one another meet with too much friction, which expresses itself in unpleasant noises and sometimes even in vibrations.

[62. (01 & 669. 1]

New materials will cut locomotive repair costs [1],

by C. E. BARBA,
Mechanical Engineer, Boston and Maine Railroad.

(Railway Mechanical Engineer.)

About the last word in the steel world are the special steels that have been produced to be used in applying the super-hard cases that are built up by heating the finished parts in ammonia gas. These are known as nitrided steels.

The ammonia gas dissociates into nascent nitrogen and hydrogen at about 850° F. The parts to be hardened on the surface are first formed, then heat treated, and then finished machined, ground and polished before placing in the retort for nitriding. The nascent nitrogen at 925-975° F. combines with the iron, chromium and molybdenum in the steel to form nitrides of these elements, and these nitrides form the hardest wearing surface that science has been able to develop, excluding the carbide cutting tools. It has been known as early as 1907 that nitrogen would, under certain conditions, combine with iron to form iron nitrides and that these nitride grains were exceedingly hard. But the hard cases formed were very brittle and spalled off from the base metal very easily.

Various investigators worked to bring out steels that would form tougher cases with nitrogen. The final result has been a steel containing chromium, molybdenum and aluminum. The chromium and molybdenum toughen the case and the aluminum acts as a catalyzer and stabilizer to maintain the high surface hardness. The steel is produced in four carbon ranges, running as low as 0.12 % to as high as 0.65 %.

The heat treating and machining are all completed before the hardening operation and that is conducted at low temperature—950-1 075° F.—so that distortion and subsequent grinding are eliminated and parts may be machined to very accurate dimensions or fit and then hardened with little danger of warping.

This case is the hardest and most uniform structure ever produced on steel and, as such, its great wear resistance is opening up vistas of long-lived parts for locomotives that were undreamed of before its advent.

The basic idea in the application of nitrided steels to locomotives is to select for its application those parts in which frictional wear causes heavy expense for renewals. The fundamental law in bearing metals is that if the two parts are of equal hardness they will both wear. If one is hard and the other soft, the softer will take the greater wear. In applying nitrided steel we endeavor to replace those parts which are most expensive to replace and to throw the wear over onto the least expensive part. Where a rotating shaft or pin is to be considered, it is generally cheaper to make the bushing take the wear.

The metallurgists have some long and beautiful stories about the microstructures and the effects of time, temperature and gas pressure in retorts, on the hardness, depth of case, toughness, etc., but the mechanical engineer is interested in two things only; namely : 1. Can wear be reduced without sacrificing strength ? 2. Will the increased life of

(1) Part of a paper presented at the spring meeting of the American Society of Mechanical Engineers, held at Birmingham, Ala, 20 to 23 April 1931.

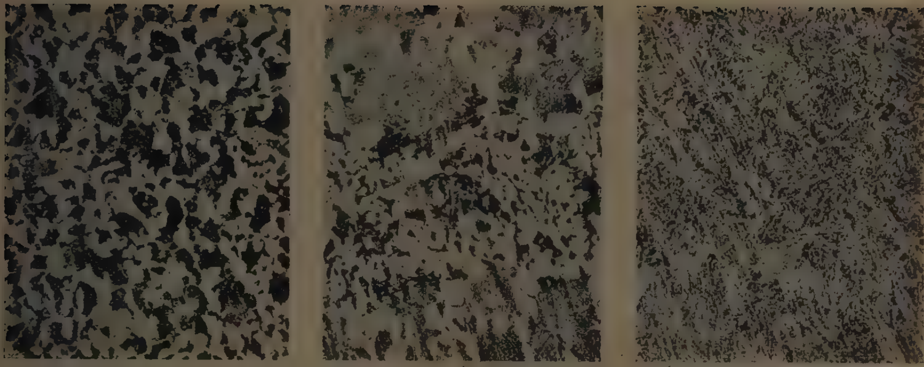


Fig. 1. — Condition as welded.

Fig. 2. — After heat-treatment.

Figs. 1 and 2. — Microstructure of welded steel cylinder bushing before nitriding Left : the plate. Center : The weld junction. Right : in the well. — 100 x (slightly reduced).

the wearing surface compensate for the higher cost of the material used ?

In preparing a wearing surface of nitrided steel we must remember that we are dealing with a surface hard enough to cut glass. Nitrided wearing surfaces must be lapped in, or polished, or glazed, before the nitriding treatment is applied. They are too hard to do much with after nitriding and, furthermore, the best wearing material is on the surface and grinding after nitriding is considered detrimental and poor practice. The selection of bearing metals to be used against nitrided surfaces is of vital importance. New and harder bearing me-

tals are being used with great success. Any application of nitrided steel should also be considered from the standpoint of the bearing metal to work with it, and the men who are building up test data and experience should be consulted.

Theoretically, all moving parts should be on a par with each other as to wear-resisting qualities, so that the weakest link, all other things being equal, in the maintenance and replacement of parts becomes the locomotives tires. The service life of wearing details, by the judicious selection of materials and improvement of design to best utilize these materials, then becomes a multiple of rather than a fractional part of the tire service life.

In order to illustrate just what can be accomplished, it might be well to start at the power end of the locomotive, namely, the cylinders, and analyze present conditions.

All locomotive cylinders are bushed with cast-iron bushings as a means of preserving the cylinder and saddle throughout the life of the locomotive. Cast-iron cylinder bushings of refined iron are undoubtedly the best practice developed through the years, but because of the unsupported piston, the bushings will wear out of round and in time must be rebored. The same thing holds true of the piston, or the bull ring, which suffers a similar wear, and must be built up or replaced with a new one.

This, however, requires dismantling of the cylinder heads, piston, etc., and the tying-up of a locomotive, and it would seem rational to assume that, if the present bushings were replaced with nitrided steel bushings, having a hardness equal to 850-950 Brinell, lapped to a very high polish, the hardness would never be disturbed, as the range of temperature is well within that used for nitriding purposes. The same holds true for piston-valve bushings. Such an installation would undoubtedly mean the elimination of future rebores and the prevention of leaks past the snap rings, thereby conserving steam.

The development of Nitralloy cylinder bushings, other than forgings, presents a most interesting problem in that plate can be used by rolling into shape and welding by the atomic hydrogen arc process, after which it may be machined to size, the bore lapped and nitrided prior to application. The steps are as follows : 1. Welding of 1 1/4-inch plate rolled into shape; 2. Heat-treatment of welded assembly; 3. Machining and nitriding of welded assembly.

This work was undertaken with the objective of forming large cylinder bushings for locomotives. After two or three trials the proper technique was developed and several welds free from blow holes were made. Chemical analyses were as follows :

		C.	Cr.	AL.
Welding wire		0.236	1.42	1.13
Plates		0.26	1.21	0.95
Weld No.	1	0.122	1.32	0.86
	2	0.178	1.30	0.86
	3	0 139	1.35	0.82

The three welds shown average about 10 points less carbon and 27 points less aluminum than the original analysis of the welding rod. The plates had about the same carbon content as the rod. The aluminum remaining in the welds is well above the 0.60 % minimum required to produce a proper case. The drop in carbon is really beneficial as the weld is more ductile than if the carbon had all been retained.

A section about 6 inches long with the weld running its full length was heat-treated, 1 700° F., water, 1 325° F., air; machined from 1 1/4 inches down to 3/4 inch, ground on both flat surfaces, and then nitrided for 42 hours at 925° F.

Hardness readings show around 850 to 900 Vickers Brinell. No difference was apparent between the readings on the welds and those made on the parent stock. Micro-section through the case

showed the same penetration on the weld as on the parent stock.

The atomic hydrogen process can be successfully used to obtain welds of special steel analysis that will produce a nitrided case identical in hardness and microstructure with the parent metal.

Dendritic structures are retained in the welds that might be broken up by forging, but they are not thought to be detrimental where the bushings are under no particular stress and are only applied to obtain a surface for resistance to wear of the piston and rings.

The success of the operation is entirely dependent on the skill of the welder and a policy of rapid fusion and application of small drops of molten metal at a time must be observed. Puddling the metal will invariably cause excessive loss of aluminum and will develop blow holes in the welds.

The utilization of special steels and the engineering problems in connection therewith are outlined for various locomotive details as a means of stimulating the art.

Piston heads and rods.

The piston head, irrespective of design, can be made of a special steel forging or casting and the rim nitrided after the grooves have been machined and lapped to dimensions, so as to eliminate groove wear from snap-ring action. This likewise applies to piston-valve parts.

Snap rings can be made of a bronze alloy, L-shaped, capable of sustaining temperatures around 900-950° F. without removing the snap, which, in contact with a highly polished surface of extra hardness, is bound to give maximum life of the piston head and rings.

The possibilities of the use of an extended piston rod should be given further consideration, as the conventional design now in use cannot be defended as one in which the best engineering principles have been incorporated.

The weight of the piston should be supported at the rear by the crosshead and at the front by a pivoted bearing for alinement which undoubtedly would minimize, or practically eliminate, the wear of piston-rod packing and the circumferential wear of the piston head and simplify the work of the snap rings, since we are starting out with a true bore lapped for accuracy and hardened. The only compression variable of the rings would be that created by the vertical and lateral wear of the crosshead.

The extended piston rod is not a new idea, but its abandonment from general use was largely due to the inability of controlling the wear between crossheads and guides, both vertical and horizontal, within limitations which would not affect the bearing surface of the extended-piston-rod crosshead at the front end of the cylinder.

The adoption of nitrided steel for piston-rods, the wearing surface of which can be hardened so as to eliminate wear, should prolong the life of the piston-rod packing, as it is certainly less expensive to replace rings than it would be to grind down piston rods.

Crosshead shoes and guides.

In the case of crosshead shoes and guides it is considered best to apply nitrided plates to the guides and to apply bearing metal to the crossheads or crosshead shoes in the form of channels which can be quickly removed and replaced at very low cost. The present practice calls for removal of worn guides and building them up with welding bronze. Then they go to the blacksmith shop to be annealed and straightened, then to the planer, the surface grinder, and finally back to the engine where they must be reset. Generally the bolt holes all have to be reamed again and new bolts fitted, so that rehabilitating a worn crosshead guide becomes an expensive job.

The proper selection of steel for controlling the wear of crosshead shoes and guides, or designing either on the basis of increasing the bearing area, thus reducing the pressure in pounds per square inch to a minimum and thereby increasing its life, should assist in reestablishing a design of paramount importance in the reduction of maintenance.

As a matter of fact, there is no reason why guides and crossheads should not be capable of operating 150 000 miles without the necessity of a take-up. The wear of guides is partially due to having a piston hanging at the end of a rod, one end of which is fastened to the crosshead, very much out of balance and ready to drop in its movement equal to the difference in diameter between the piston head and the bore of the cylinder, the tendency of which is to cock the crosshead at the end of the stroke.

Side-rod bushings.

Side-rod bushings have ever been a source of trouble due to rapid wear. The selection of bearing metals is one

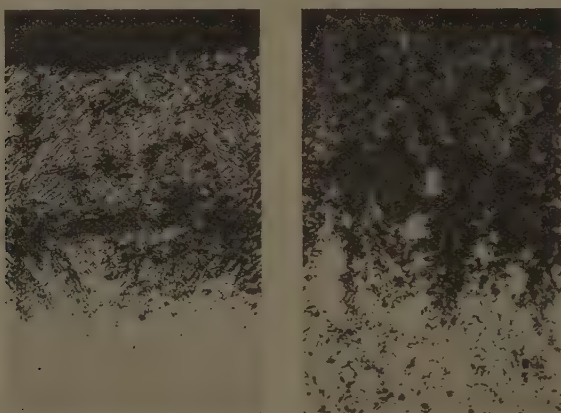


Fig. 3. — Microstructure of the nitrided case. — Left : On the weld metal. Right : On the plate surface.

of utilizing a bronze that can be subjected to a higher working temperature than is now possible and, when used in connection with hardened crank pins, the wearing qualities can be increased, while seizure from operating temperatures and lack of proper lubrication will be practically eliminated.

Driving boxes.

Driving boxes, as designed at present, have not sufficient sustaining qualities for the application of a pressed-in crown brass, due largely to the limitations of present pedestal spacings, all of which can be modified by the elimination of shoes and wedges.

The use of present shoes and wedges is to take care of wear of the box, liners and shoes. To minimize this wear so that they will operate between shoppings within prescribed tolerances will necessitate the use of nitrided-steel wearing surfaces. Furthermore, the use of shoes

and wedges often creates improper transverse alinement, or parallelism, of driving axles, which, if eliminated, simplifies shop operation.

The design of box could, therefore, be modified by utilizing a portion of the thickness of the present shoes, wedges and floating liners by increasing the overall width. The wearing surface of the driving boxes and pedestal fit should be a flanged liner on both pedestal and box, made of manganese steel not less than 3/16 inch thick, the life of which should not be less than 200 000 miles.

Such a design would permit the use of a crown brass more rugged in design, which can be applied without pressure so that replacement is simplified and present costs minimized.

The ever-increasing lateral wear between driving box and wheel hub can practically be eliminated by the application of nitrided-steel liners, inserted in a recess in the hub of the driving wheel and welded on the periphery, in conjunction with a similar liner inserted in the face of the driving box, the hardness of which should be approximately 850-950 Brinell. Such liners, when coming in contact, either lubricated or non-lubricated, should be good for 200 000 miles.

Driving axles.

Locomotive axle failures are an increasing difficulty and expense. There are two types of failures. The one type is a fatigue crack starting in the fillet next to the wheel center. This type of failure generally starts in a rough finished fillet, or where the brass cuts a shoulder in the fillet. Anything that will produce a notch effect on the bearing surface of the axle will start a crack that will progress until the heavy starting torque and the bending moment on the axle, when the crank pin passes over the center, will produce enough strain to rupture the remaining portion. The second type in one of hot boxes which produce thermal checks on the journal surface, and these are starting points for progressive fractures.

Anything that would relieve the axle of the duty of supplying a wearing surface in addition to resisting the high fiber stress due to flange pressures would prolong the life of the axle. Axle bushings of nitrided steel, shrunk onto the axle, are now being tried out in testing laboratories. There is no reason why a pair of wheels, when mounted, should have journals re-turned to true up worn surfaces. A pair of driving-wheel centers properly mounted on a nitrided steel axle, having journals hardened or bushed, together with nitrided-steel crank pins properly quartered, should never be disturbed except for the replacement of tires.

Locomotive links and motion parts.

Locomotive links, the design of which has not been altered for many years, require continual regrinding due to the movement of the block. If made of nitrided steel and used in conjunction with a hardened block having a Brinell of 500-600, regrinding can be eliminated.

When one considers the importance of proper steam distribution, the ideal valve motion, irrespective of type, would be one in which all parts would never require any adjustment due to wear of pins and bushings. Here again the utilization of nitrided steels will go a long way toward establishing a greater degree of accuracy and more perfect steam distribution than is now possible.

Spring rigging.

The function of a spring rigging is to take care of track irregularities, tire wear, etc., and the proper distribution of weights on the running gear. Consequently spring design and materials selected play a most important part. The gradual elimination of the present ellip-

tic spring and the adoption of a different form of spring suspension than used at present, together with a re-design of the hangers that will eliminate wear and preserve the alinement of the locomotive, irrespective of tire or crown-brass wear, is in order. There is no good reason why the designing and manufacturing of locomotive springs should not follow automotive practice.

From the foregoing it will be observed that there is opportunity for radical departures from present-day practice and, unless those directly interested in engineering principles covering locomotive design change their viewpoint in the utilization of ideas and materials used so successfully in other lines of industry, we shall be unable to profit from the progress which has been made in those industries for our own use, as a possible means of decreasing maintenance.

Wheel centers.

The design of wheel centers and the character of steel used is a subject that requires more research.

Taking a concrete example, specifications require that all steel castings must be annealed so as to relieve strains introduced during cooling. This seems to be a universal practice. On the other hand, we take a wheel center, press it on an axle, introducing compressive stresses in the hub which may or may not be transferred into the spokes, and then press in a crank pin, the stresses of which go into the hub and spokes, and finally shrink on a tire, which not alone compresses the rim, but likewise the spokes, the result of which is a wheel center having all sorts of internal stresses. As a matter of fact, it is stressed far in excess of any stresses that may have been introduced originally during the casting and cooling process. We still insist that those stresses must be removed but nothing is said of the stresses of the completed article.

It would seem, therefore, that tests are in order calling for the application of extensometers on spokes, hubs and rims, covering all of these operations for the determination of stresses, or for the possible redistribution of metal, if it is found necessary, to take care of those stresses, which should assist in furthering the life of our present wheel centers.

Locomotive tires.

Present practice in locomotive tires, or for that matter any tired wheel, irrespective of size, calls for three grades of material; namely, *A*, *B* and *C*; *A* used in passenger service, *B* freight service, and *C* switcher service, representing graduated increases in carbon content and, consequently, hardness to resist wear. These materials have undoubtedly stood the acid test for a good many years. There is a question, however, among metallurgical engineers as to whether the railroad engineering profession has not overstepped its limits with respect to wheel loads, taking into consideration the small contact area that a tire makes with the rail.

This subject undoubtedly covers a great many years of intensive research, primarily metallurgical, but limited in engineering considerations, as mass has not been considered an important item where fluidity of metal (due to great pressures) in concerned. It is believed that better results could be obtained by increasing the tickness of a tire over the conventional thickness so as to have more backbone in assisting the transmission of weight to the rail and, naturally, giving more life on account of this additional thicknees.

Toughness in tire materials is a determinant in wear resistance, but, unfortunately, in ordinary tire steels, toughness is only obtainable by increasing the carbon content, which produces a material of improved wearing characteristics but of a hardness with increased ten-

dency to fracture under shock stress. However, with the change in speeds of freight locomotives narrowing the gap differentiating from passenger operating characteristics, it is felt that we can use the harder *B*-grade tires for all *A*-grade purposes.

Further, to meet the demands of maximum hardness and high tensile and elastic limit to resist flowing tendencies under heavy wheel loads, and to advance the service life of tires to meet the possibilities of increased life in other wearing parts, considerable attention should be given to the development of alloy steel for tires. Tires are now being used with nickel and manganese content, giving harder wearing surfaces without the brittle tool-steel characteristics of high-carbon tires. The surface as yet, however, has only been scratched.

Firebox steels.

The major problem in the selection of steel for fireboxes is one of temperature and requires a steel that will resist whatever the maximum temperatures of a firebox may be (under various operating conditions) so as not to break down the structure of the material. This, in itself, is quite a large contract, and various grades of steel have been tried with varying results, but not with sufficient assurance that their adoption would eliminate all of the troubles now experienced. It would, therefore, appear that one of the major problems is one of design in which the flow of gases over the arch will be similar to the flow of water over a dam; that is, straight line movement. The influence of the draft-producing medium at the center of the smokebox undoubtedly causes a stream flow of the gases converging toward that center, thus materially lessening gas flow and heat transfer away from the center of the tube sheet, all of which has a tendency toward creating hot spots in the firebox, rather than maintaining a uniform temperature throughout.

Furthermore, the percentage of carbon in firebox sheets is believed to require careful consideration and should be a minimum—excluded if possible, as any carbon content unnaturally must have a hardening influence when heated and cooled, sometimes abruptly, under the present operating conditions. Experiments are now being conducted with a brick side-wall lining to determine the effect of controlling the rapidity of temperature change by a more uniform heat distribution.

Grates.

The design of grates and the selection of a material that will resist heat at higher temperatures than any material now in use would be a step in the right direction, although grates may be so designed that their section may be sufficient to reduce the temperature at the top of the grate well within the confines of the critical range of the material by the net area and distribution of the air inlets. This, however, requires some experimentation for determination.

Staybolts.

Design likewise should more fully enter into the staybolt situation, rather than the mere selection of better material, or type of material. The entire firebox, were it not for the attachment of the firebox sheets to the mud ring, is a floating chamber supported by staybolts. The water legs, increasing in width from the mud ring to the crown sheet, with the space between the crown sheet and the wrapper sheet being in excess of that of the widest width of the leg, calls for staybolts of varying lengths, but practically all having the same diameter.

Considering the staybolt as a cantilever beam, that is, by assuming that there is no movement of the wrapper sheet relative to the firebox sheet, the number of pounds required to deflect a long stay-

bolt is naturally less than that of a short staybolt having the same diameter. Consequently there are zones in each firebox side sheet that are restrained, due to the constant force of expansion at uniform temperatures, by the shorter bolts not deflecting the same amount as the longer ones which produce stresses within the sheets, that in time will produce hair line cracks around staybolts. Eventually failures result.

As a suggestion, or at least as something to think about, why not divide the application of staybolts in a vertical plane into horizontal zones and design the staybolts in certain zones according to their length by assigning diameters so that all staybolts may deflect uniformly, assuming that the expansion force is constant.

Tank steel.

The selection of materials likewise can be extended to the use of tank sheets instead of present open-hearth steel. Even though copper-bearing steels are being used to a very large extent, other materials can be procured in which corrosion can be eliminated and, therefore, the initial cost can be justified, considering the elimination of tank repairs which are now prevalent.

It also would be helpful in designing the slope sheets to have the joints made on the vertical sides instead of where the slope sheet coincides with the side sheets, and providing at least a 6-inch or 9-inch radius in the corners. This would not only assist the coal in passing to the conveyor more freely, but would prevent the corrosive elements from entering the joints made in present-day construction.

Design as important as materials.

The utilization of alloy steels is not the only attack on maintenance. The best of engineering practices and industrial arts must be drawn upon in the development of integral designs to replace built-up assemblies for the purpose of realizing maximum strength and reducing wear and tear at joinings. This is strikingly exemplified in the design of the integral bed frame for a locomotive foundation.

This represents a radical and bold departure from past practices, yet experience has now been sufficient amply to justify the design.

It is believed that the money expended for the initial cost of using nitrided steel and the other special materials referred to as a means of decreasing maintenance will be more than justified in the subsequent reduction in maintenance cost and lesser loss of service of locomotives held for maintenance. An accounting system should, if necessary be developed by which the cost of repairs or replacements of individual details of each class of locomotive can be recorded for easy reference and to serve as a source of reliable reference data to substantiate the foregoing belief.

In conclusion, it might be well to emphasize the wonderful possibilities in design in conjunction with the utilization of high-grade material, which the railroad engineering profession should consider, as the selection of steels is only a portion of the problem.

CURRENT PRACTICE.

[625. 232 (.4)]

All-metall rolling stock of the International Sleeping Car Company.

The International Sleeping Car Company put into service, rather over a year ago, seventy sleeping cars and forty restaurant cars built entirely of steel. It has under construction in different works in Europe a further twenty-five sleeping cars and twenty restaurant cars of this type which will be in service in a few months' time.

1. — Sleeping cars.

The vehicles previous to this new type had an all-metal body and roof but the interior decoration consisted of plywood panels decorated by marquetry work. Some of the partitions and the doors were also made of wood.

This is no longer the case in the new sleeping cars we are considering, as in their construction wood has been entirely eliminated. The interior partitions, the doors, the wash-stands, the tables, and the ceilings are all made from sheet steel.

Particular care has been taken to insulate these vehicles as regards noise as well as against outside differences of temperature.

The insides of the body sides and of the roof have been lined with mattresses (salamander-compressed cork) carefully fitted to the steel plates.

The inside partitions, all of which are hollow, have been lined on their inside faces with backing canvas to prevent vibration.

The floor is made of dovetailed sheet covered with a thick layer of compressed cork, over which is laid in turn linoleum and carpet.

The whole of these precautions which have been studied with the greatest care has resulted in the vehicles being absolutely silent.

These carriages are carried on monobloc bogies of cast steel of the Sleeping Car Company's standard type; the good riding qualities of these bogies have been recognised for a long time.

The interior decoration is obtained by means of cellulose lacquers applied directly to the steel sheets.

This very modern decoration due to a decorative artist of great repute, Mr. Prou, consists of bands of colour on colour and bands of creams. The ceiling and the upper part of the divisions in the compartments are painted white.

All metal fittings : luggage racks, coat-hooks, locks, handles, etc., are chromium plated.

The seats and the arm-rests are covered with « tobacco » coloured velvet pile.

The floor of the compartments and of the corridor is covered with a carpet of modern design of red, beige, and garnet colour.

The partitions and doors have few mouldings and these are thin, so as to facilitate cleaning.

The appearance of this decoration is both light and cheerful.

The sleeping cars of this pattern have the following accommodation.

1. 11 two-berth compartments, five of which intercommunicate by means of double doors in the Z-shaped partitions separating them.

2. Two toilets at the ends of the body;

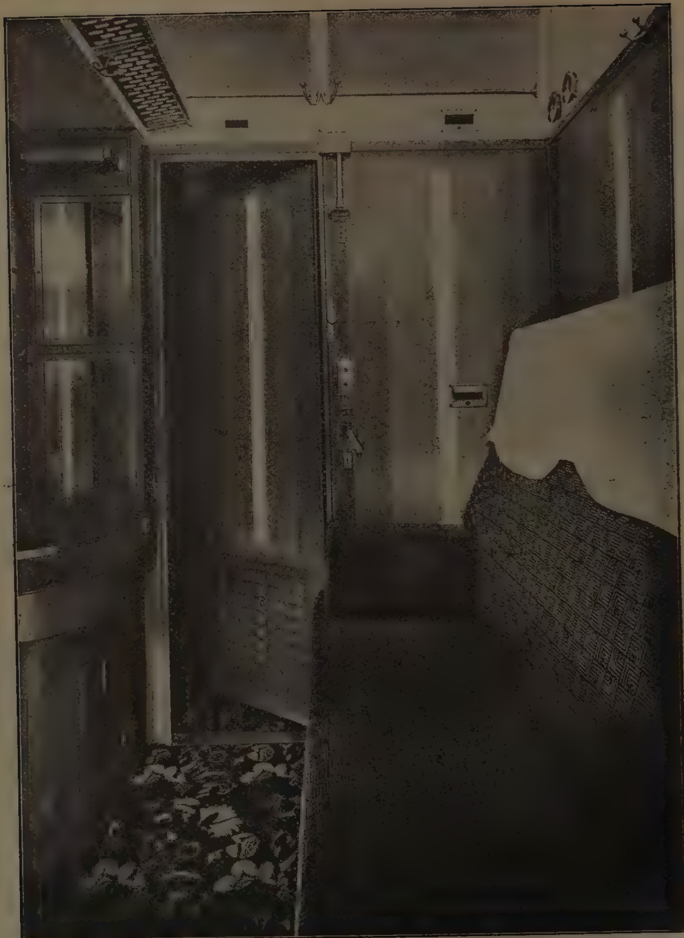
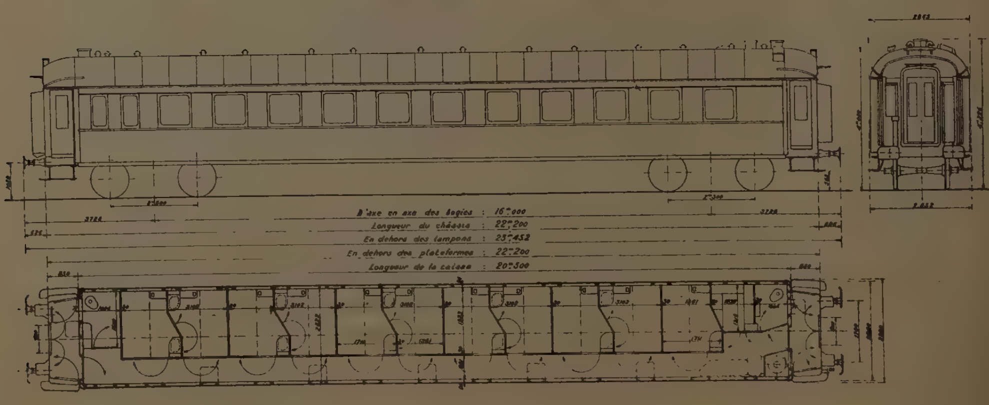


Fig. 1.



D'axe en axe des bogies : 16m000
Longueur du châssis : 22m200
En dehors des tampons : 23m452
En dehors des plateformes : 22m200
Longueur de la caisse : 20m500

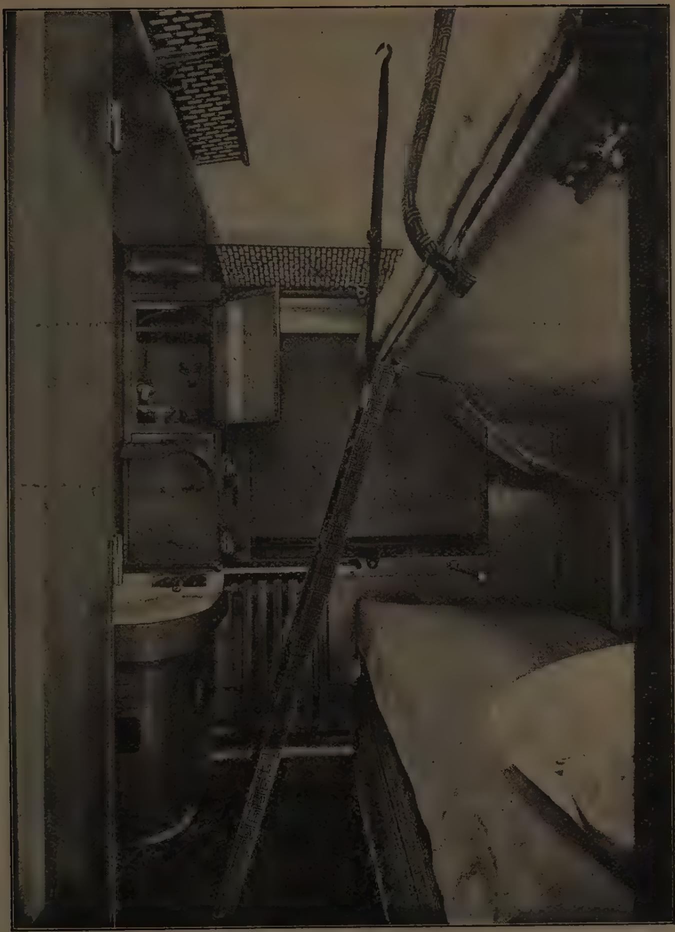


Fig. 3.

Fig. 4. — Restaurant car.

Explanation of French terms en figs. 2 ánd 4 :

D'axe en axe des bogies = From center to center of bogies. — Longueur des châssis = Length of frame. — En dehors des tampons = Over buffers (overall length). — En dehors des plateformes = Outside end corridors. — Longueur de la caisse = Length of body. — Cuisine = Kitchen. — Office = Pantry — Places = Seats.

3. Pantry;

4. Heating chamber;

5. Corridor running along the berths to which it gives access;

6. Two vestibule ends.

Compartments. — The equipment of each compartment includes :

1. Convertible seat with back adjustable for day and night use. In the day position, this seat with its back set back against the partition forms a seat and gives the berth the appearance of a coupé compartment (see fig. 1).

In the night position the seat is pulled out a little from the partition and acts as a mattress for the lower bed. The back is pulled up and held in the horizontal position by catches provided for this purpose and then forms the upper bed.

These two mattresses are made up of very light small springs in cases covered over with a layer of hair.

Each bed has furthermore a hair and wool overlay mattress, a pair of sheets, pillows, and two red blankets.

2. A wash basin in one of the angles of the Z-shaped partition. This fitting has a hinged lid which covers the basin and can be used as a table when the basin is not being used.

Above this fitting there is a small cupboard containing a water carafe, two glasses and towels. The door of this cupboard has a mirror on each side and is so fitted that it can be held open in several positions to suit the convenience of the passengers.

The wash basin has chromium plated taps giving cold and hot water as desired by the passenger, even in summer.

3. A radiator above which is a small table with a built-in ash tray for the use of smokers.

4. Two luggage racks, hat hooks, coat hooks, bottle holders, and glass holders.

A ladder to give access to the upper berth, covered with velvet pile is also provided in each compartment. In addition two large spaces for luggage have been formed in each compartment above the corridor, access being given through openings in the partition.

In each compartment there is a drop light, one metre (3 ft. 3 3/8 in.) wide, fitted with the Hera balancing gear and easily operated by a small handle.

These windows are also provided with blinds and curtains.

Electric lighting. — Each compartment is lighted by a ceiling fitting consisting of three 10-candle power lamps, without shades, all three being lighted at the same time. A 6-candle power reading-lamp fitted in the corridor partition is provided for the use of each of the two passengers when lying down. A night-light protected by a blue-shade is let in over the compartment door. All these lights are controlled by switches conveniently placed under the passenger's control when sitting or lying down. Finally a 6-candle power tubular lamp is fitted above the mirror of the wash basin and is controlled by a switch just below it.

Heating. — Each compartment is heated by hot water radiators under the passenger's control. The corridor is heated by pipes running along the whole length at the bottom and the top of the body side.

The water in these pipes and in the radiators is heated to about 90° C. (194°

F.) either by a coal-burning boiler placed in the heating compartment or by a steam heated coal heater in this compartment.

Ventilation. — The ventilation is obtained by providing in each compartment a louvre ventilator above the window and an extractor in the roof, controlled by a small lever on the longitudinal partition.

In addition, the doors of the compartments have been fitted at the bottom with narrow louvres opening into the corridor.

The corridor is ventilated by hand-operated extractors in the roof.

Pantry. — A small pantry is provided in these sleeping cars so that passengers may be given breakfast in the morning in addition to being served with tea and mineral waters.

This small pantry has been fitted with a charcoal stove, an electric kettle, a drinking water container, a sink and many cupboards for linen, china, and silver.

Toilets. — A toilet is provided at each end of the body. The ceiling and walls are made of sheet steel. The ceilings and the top of the walls are painted white, cellulose lacquers being used. The lower part of the walls is painted grey.

The floor is covered with silver and cream mosaic.

Each of these two toilets is ventilated by a drop-light window 490 mm. (19 1/4 inches) wide fitted with embossed glass. A roof extractor without cover is also fitted.

One of the toilets is fitted with a silver-plated wash basin.

These all-steel sleeping cars are extremely strong and provide the maximum degree of safety against accident.

They weigh 54 tons and are 22.200 m. (72 ft. 10 in.) long over vestibules (see fig. 2).

Passengers in possession of a 2nd-class railway ticket and paying the 2nd class sleeping car supplement can reserve places in these cars. Under these conditions each compartment is occupied by two passengers, one of whom uses the lower berth during the night, and the other passenger the upper, the bed of which is formed by the seat back raised to its horizontal position (see fig. 3).

Passengers with a first class railway ticket paying the sleeping car 1st class supplement are each given a compartment to themselves. At night the under mattress is formed by the seat. The back is set back closer against the partition and its top roll is lifted upwards to reduce the space taken up. A white linen cover is put over a large part of the back at the head of the bed at night.

In the day time white antimacassers with an embroidered WL monogram are provided on the backs.

2. — Restaurant cars.

The interior arrangement of the new restaurant cars includes (see fig. 4):

1. A kitchen, 3 m. (9 ft. 10 1/8 in.) long and about 2 m. (6 ft. 6 3/4 in.) wide, ventilated by means of 3 windows fitted with glass louvres;

2. A pantry of about the same size as the kitchen, ventilated by two windows, one fitted with glass louvres;

3. Side corridor alongside the kitchen and pantry;

4. Large smoking section 7 m. (22 ft. 11 5/8 in.) long with 8 tables to seat 4.

These sections are ventilated by 6 drop lights of which four are fitted with glass louvres;

5. Non-smoking section 5.286 m. (17 ft. 4 in.) long with six tables seating 4 each.

This section is ventilated by 6 drop lights of which four are fitted with glass louvres;

6. A large cupboard, the top being for clean linen;

7. A chamber in which the heating appliances are fitted;

8. A pantry arranged as a wine cellar;

9. A heating chamber;

10. Two vestibules at the ends of the vehicles.

In these new restaurant cars the whole of the inside linings, the partitions, the doors, and the seat frames are made from sheet steel.

The decoration, as in the case of the sleeping cars, is obtained by cellulose lacquers applied directly to the sheet steel.

This decoration is much like that used on the all-metal sleeping cars. It consists of bands of three different shades of green, with bands of cream.

The lower part of the body side of the dining sections is painted aluminium colour.

All brass work : luggage racks, lamp holders, handles, locks, etc., is chromium plated.

The ceilings are painted white.

The floor in the two sections is wholly covered with linoleum and a carpet strip has been laid between the tables.

The seats and backs are stuffed with hair and covered with goat skin of tobacco colour.

Lighting. — The two sections are lighted by 14 electric lamps with closed-in tulip shaped shades in cristal let into the ceiling round the raised portion, and by 28 lamps arranged in pairs above each table, between the luggage rack and the window.

Heating. — The two sections are heated by hot water circulating through three pipes running right along the lower part of both body sides.

The water in these pipes is heated in the same way as on the sleeping cars described above.

Ventilation. — In addition to the glass louvre ventilators above the eight windows, two large electric exhausters fitted with suitable covers have been arranged in the roof of each section.

MISCELLANEOUS INFORMATION.

[625 .143.3 (.73) & 625. 245 (.73)]

Transverse fissures can now be located.

We deem it interesting to reproduce, hereafter, as a sequence to an article which appeared under the above heading in the March 1929 number of the *Bulletin of the Railway Congress*, a series of twelve photographs of typical internal transverse fissures found in rails.

These illustrations are taken from *Bulletin* No. 57 — *The Sperry detector car* — of the Sperry Products, Inc., of Brooklyn and Chicago, U. S. A.

Fig. 1. — Area of fissure 1.5 % of rail head. Typical example of small fissure which seems to have started from a small longitudinal seam. This fissure is considerably smaller than the nucleus of other fissures shown.

Fig. 2. — Aera of fissure 5.3 %. Typical example of fissure developing from a large nucleus. The Sperry car has even located and recorded nuclei only, before development of the fissure itself.

Fig. 3. — Area of fissure 23.6 %. This is a medium size fissure so often found by Sperry cars. Fissures of this size or even smaller have caused faillure of rail in track.

Fig. 4. — Area of fissure 50.4 %. A very dangerous and common type of transverse fissure which, as immediately noticed, has already weakened the rail head by one-half.

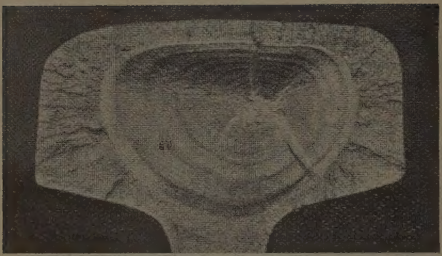


Fig. 5. — Area of fissure 53.4 %. A still larger fissure and one very interesting because of clearly defined rings which unquestionably indicate definite periods of growth.

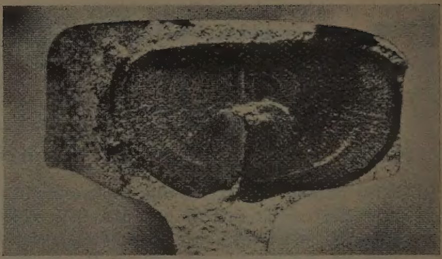


Fig. 6. — Area of fissure 63.9 %. Notice fissure has reached the surface and therefore has started to discolor. An exceedingly dangerous defect and surprising that rail had not already broken.

Fig. 7. — Example of horizontal fissure so commonly found by Sperry detector cars.

Fig. 8. — Horizontal fissure in a mild manganese rail. Note smooth surface of rail at fissure.

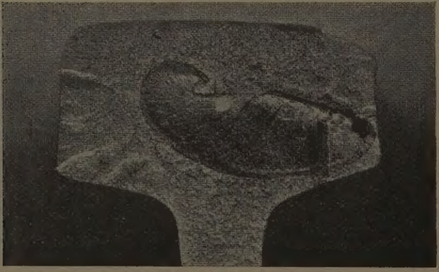


Fig. 9. — Area of fissure 32 %. Typical compound fissure. Such fissures indicate that split heads, horizontal fissures and other longitudinal seams are potential nuclei for transverse fatigue failures.

Fig. 10. — Area of fissure 65 %. Typical example of a very large and dangerous transverse fissure which has developed from a horizontal seam. Note the very small section of good rail left.

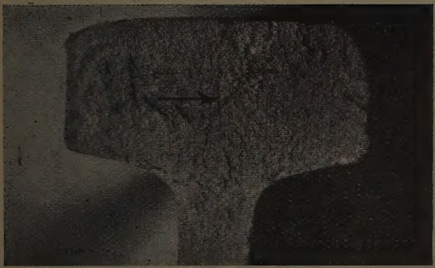


Fig. 11. — Typical split head in early stages of development detected by a Sperry detector car. Rail was broken and the accuracy of the Sperry method again proved, even for such very small cracks.

Fig. 12. — An interesting combination of pipe and transverse fissure at the same location in a rail. Notice the split head is very much larger and more dangerous than in the illustration to the left.